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PRODUCT ENGINEERING

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How to Design Worm Gearing
Plastics—Motors—Bearings
Second Series of Case Histories

•

Mechanical Design • Product Development

McGRAW-HILL PUBLISHING COMPANY, INC.

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PRODUCT ENGINEERING

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IN THIS ISSUE

Mechanical Design — Three major articles on mechanical design feature this number. The article by C. H. Grill is a clear cut presentation showing how an engineer who has specialized for many years on the design of worm gears, goes about it.

Allowable stresses and the special requirements when designing spur gears to be made of laminated plastics, are given by H. M. Richardson of the General Electric Company.

And in this issue we present the first of two articles by F. A. Firnhaber, designing engineer of the Landis Machine Company, on anti-friction bearing mountings for medium-speed heavy-duty spindles.

Motors—A two-page article setting forth how some chief engineers regard independent motors for driving the different units of a machine.

Product Design—Beginning the 1934 series of Case Histories wherein we delve deeper into the reasons for the designs and why they were successful.

COMING ISSUE

Having presented the question of hydraulic versus mechanical feeds, we follow up in the February number with an article by James A. Hall who gives practical methods for calculating gear trains.

Another feature article will be on the application of motors, electric controls, solenoid brakes and magnetic clutches to a machine tool.

Other coming articles include motor applications, critical speed of shafts, designing for molded plastics and practical considerations in the design of die-castings. Also comprehensive articles on types of light-weight construction with reference to the materials used and design details.

GEORGE F. NORDENHOLT, *Managing Editor* — H. W. CARSON, *Assistant Editor* — KENNETH H. CONDIT, *Editor and Publishing Director* — LEONARD CHURCH, *Cleveland* — PAUL WOOTON, *Washington*

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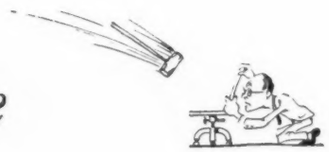
Intimate Correspondence



Color

PRODUCT ENGINEERING

NEW YORK

Why has there been so little information on the use of color to add attractiveness to the product in recent issues of *Product Engineering?* You ran some excellent articles a year or so ago. Did you exhaust the subject?

WILSON R. HENRY,
International Products Co.

OF course we have not exhausted the subject. As a matter of fact there is still much to be learned even by the experts in color application. But the average citizen has had too many other things to worry about these last few months what with the multitudinous alphabetical activities in Washington and the leanness of the personal pocketbook. Clear evidence of the trend is given by the color preference of automobile buyers. The last report showed a continuing demand for black, always an indication that the buyer expects to get long service out of what he buys and is therefore picking a color that is least likely to tire him.

We venture to predict that the forthcoming exposition of industrial art in New York (see News page) will be an awakener of interest in color aided and abetted by a more optimistic attitude of the observer.

Hic!

PRODUCT ENGINEERING

NEW YORK

Please see page 413 of your November issue and favor me with the name and address of the maker of the Dumore electric cocktail shaker.

WILLIAM MCCARTHY

CAN it be that we have here a case of conscientious conviviality? A good citizen who has been waiting for repeal? Probably not. The chances are that modernization and mechanization have scored another hit, just as they always do when rightly used by a skillful engineer.

For Reference

PRODUCT ENGINEERING

NEW YORK

We are very much interested in one of your articles in the December issue of *Product Engineering,* having reference to the electrical and mechanical properties of hard rubber, vulcanized fiber and laminated Bakelite. We wish to secure an extra copy of this issue and would therefore appreciate your mailing one to the attention of the writer.

S. W. PLACE,
Synthane Corporation

THIS recognition of the value of a *Product Engineering* article by a leader in the field touched by the article is most gratifying. We have the conviction that such articles giving practical working data on materials are as useful as any we can print, but it is good news when such support for the conviction is forthcoming.

On the Other Hand

PRODUCT ENGINEERING

NEW YORK

The enclosed is for your records in case you decide to publish another table of corrosion resisting alloys. One or two other very fine metals are also omitted from your listing which should have been included if you intend to cover the field.

Tables of this nature, published from time to time, are of inestimable value to engineers in general. We would suggest tables concerning sleeve types as well as roller type bearings and their uses, and, for that matter, for unlimited specific applications throughout industry where a series of metals give differing results under differing conditions.

Trusting you will accept the above as a reader's criticism.

GEORGE K. DREHER,
Ampco Metal, Inc.

WE are equally pleased to get criticisms like this one of Mr. Dreher's. Particularly when they end up with requests for specific material. The more the merrier.

Commendation

PRODUCT ENGINEERING

NEW YORK

We have gone over the article in the recent issue of *Product Engineering* and commend you on the manner in which the subject of photoelectric control has been handled. Generally speaking, the past few years have seen a tremendous amount of unwarranted, or one might even say illegitimate, publicity given to photoelectric apparatus through newspapers and periodicals who were looking for something unusual to write about. In your article, however, we note that you have discussed applications which we assume you have actually seen in operation or on which you have received definite information as to their feasibility. So many discussions of photoelectric equipment are so visionary that they serve only to create false ideas and cause great confusion as to what is the legitimate field for photoelectric equipment.

For articles such as you are presenting, I presume that the prime requisite for any installation which you plan to discuss is the question of practicability, and we concur whole-heartedly in this.

We would like very much to see the rest of the articles when they are printed.

A. G. BRADT,
G-M Laboratories, Inc.

COMMENDATION from those in a position to know is doubly welcome. Mr. Bradt is correct, the examples given in the article are all staff-checked.

Needs It

PRODUCT ENGINEERING

NEW YORK

The type of my work has been changed recently and now I need *Product Engineering* more than ever before. P.E. was probably a factor in my securing this advancement and I am sure that it will help me handle the job.

HARRY I. HAZZARD

IN these days keeping up with the times is essential for every engineer. Reading such periodicals as *Product Engineering* is one of the best ways to do it.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

JANUARY, 1934

McGRAW-HILL PUBLISHING CO., INC. VOLUME FIVE, NUMBER ONE

What 1933 Has Promised 1934

LOOKING back over the year 1933, one is impressed by the epoch-making events that took place. Uppermost in the minds of all of us is the crisis through which this nation passed at the time its administration was taken over by the new chief executive, who was voted into office by a dissatisfied electorate. Immediately following his inaugural, new legislation was passed with great rapidity. And in almost every instance the new laws were enacted for the primary purpose of sweeping out of existence the social, financial and political conditions that had been favorable to the amassing of huge individual fortunes at the expense of poverty and privation for the masses. For engineers the most important thing was that the old ballyhoo of "technological unemployment" yielded to a sane recognition of the fact that it was finance and government that had failed to progress in step with science and engineering.

In the second half of 1933 began the great reawakening of activity. The opening of the Century of Progress Fair seems to have been the signal. Here millions of people saw science in action and beheld the wonders of engineering. The great consuming public received an education that has made them more discriminating in the fine points of the products they might buy and their desire to possess or share in the latest offerings of new products has been increased greatly.

History repeating itself, railroad purchases initiated the resumption of activities in the capital goods industries. This time, however, the business was not solely the replacement of existing designs. A large portion of it was the purchase of entirely new products. Air-conditioning equipment, light-weight, high-speed trains, and ultra-modern steam and electric locomotives make up a considerable share of the railroad orders placed recently.

What is the significance of all this to product engineers? There are three outstanding promises. Socially, the masses are being educated in a wholesale fashion to a keen appreciation of the things that science and engineering can give them, and their desire for these things is being greatly intensified. Politically, the aristocracy of wealth has been forced to yield to a new aristocracy based on brains and accomplishments. With the expected greater distribution of wealth and the strongly cultivated desires for the "good things" of life, there will be a greatly increased demand for products that bespeak the best that art and industry can offer at prices within reach of the "average." And the machinery of production will have to be tuned to make it possible to manufacture at a low cost the creations of the product engineer. Economically, all this signifies that in the year 1934 we shall have to leaf back through the pages of history to find the depression. And those who fail to heed the signs of the times and do not design their products accordingly will likewise find themselves in the pages of history, sitting on a dusty shelf.

Volume 5

WITH this number *Product Engineering* starts its fifth year and its fifth volume. Founded just at the beginning of the depression it has won recognition from its readers of which the editors are proud, and support from subscribers and advertisers for which they are grateful. In its fifth volume it will continue to serve in every way possible the men in charge of mechanical design and product development. The editorial staff is made up as follows: Kenneth H. Condit, editor; George F. Nordenholt, managing editor; Robert W. Carson, assistant editor; Paul Wooton, Washington; Leonard Church, Cleveland; N. J. Van Ness.

Practical Data and Formulas for the Design of Worm Gears—I

C. H. GRILL

Formulas for efficiency, loads, strength and proportioning, material specifications, and data that have been used as the basis for the design of hundreds of worm drives that proved their worth. The author was formerly chief engineer of a leading gear manufacturing company

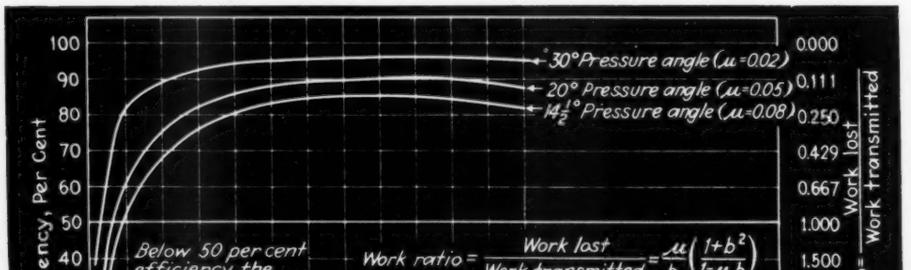


Fig. 1 — Efficiency curves of worm gear drives for various pressure angles, helix angles and coefficients of friction

WHEN a worm gear drive is to be designed, the average engineer is confronted with much conflicting data and information on load carrying ability, over-all efficiency, and factors affecting the general suitability of the worm drive for his particular purpose. Handbook information differs with each author and gives varying results. Neither will such results compare favorably with ratings contained in the catalogs of reputable firms who specialize in the design of worm gears. Naturally, any article on this subject must deviate in part at least from the data and principles set forth by recognized authorities. This article will not be a discourse on theoretical considerations. They are omitted entirely. The recommendations and suggested practices set forth here are those that have been incorporated into hundreds of successful drives over a period of years and drives that are "standing ing up" under all conditions and in all classes of service.

Only the common helical type of worm with straight sided threads will be considered. This type of worm

TABLE I

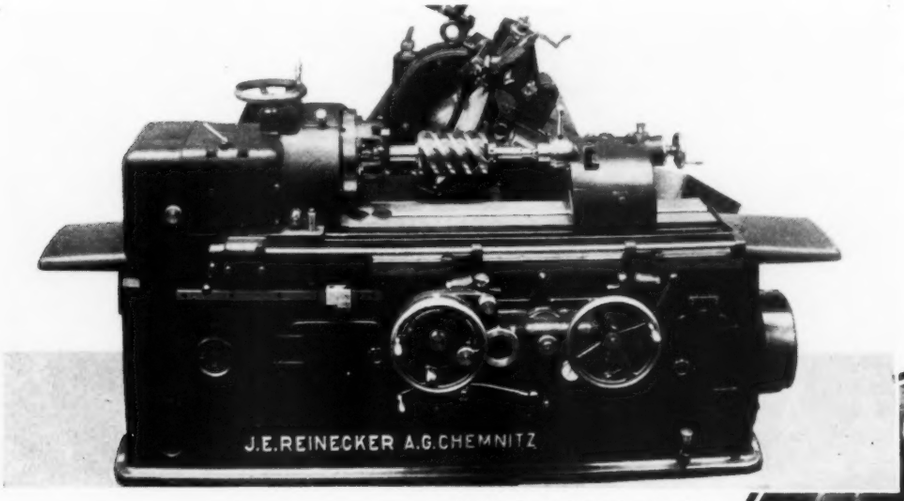
Pitch Diameter Increments to Eliminate Tooth Interference

Diametral increments to be added to the theoretical pitch diameters of worm gears having 30 teeth or less. These values are correct for 1-in. circular pitch, having an addendum = 0.3183 in.; for other pitches multiply by the circular pitch, and for non-standard addenda take proportionate values

No. of Teeth in Worm Gear	Pressure Angle of Hob		
	14½ Deg.	20 Deg.	30 Deg.
30	0.0352 in.	0.0183 in.	——
29	0.0551 "	0.0328 "	0.0106 in.
28	0.0751 "	0.0482 "	0.0212 "
27	0.0955 "	0.0637 "	0.0318 "
26	0.1152 in.	0.0786 in.	0.0424 in.
25	0.1353 "	0.0941 "	0.0530 "
24	0.1553 "	0.1095 "	0.0636 "
23	0.1754 "	0.1248 "	——
22	0.1954 in.	0.1401 in.	——
21	0.2155 "	0.1555 "	——
20	0.2355 "	0.1707 "	——
19	0.2556 "	——	——
18	0.2769 in.	——	——
17	0.2957 "	——	——
16	0.3158 "	——	——
15	0.3358 "	——	——

and its mating gear have passed through various stages of development. Thanks to the work of Bruce, Lanchester, Thomas and other early investigators, the question of worm gear efficiency is no longer a matter of doubt. Carefully conducted tests show that efficiencies as high as 96.1 per cent have been obtained repeatedly and efficiencies of 90 per cent are common. These are over-all efficiencies, including all losses in the worm and gear itself as well as the losses caused by bearing friction. But the highest efficiencies are attainable only by careful attention to the design, the use of the highest grade of materials, accurate manufacture and the mounting of both worm and gear on anti-friction bearings with rigid supports. In addition, adequate and constant lubrication must be provided.

Properly designed worm drives will give efficiencies that will conform closely to those shown on the curves in Fig. 1. These curves are based on Bruce's formula and correspond to coefficients of friction of 0.02, 0.05, and 0.08 for pressure angles of 30 deg., 20 deg., and 14½ deg., respectively. It is to be noted that the highest observed efficiencies were obtained with a pressure angle of 30 deg., which explains the adoption of this angle by manufacturers who make a specialty of standardized worm gear drives.



Courtesy George Scherr Company, Inc.

Fig. 2 — For heavy loads or high speeds it is extremely important that the worm threads have the highest attainable accuracy

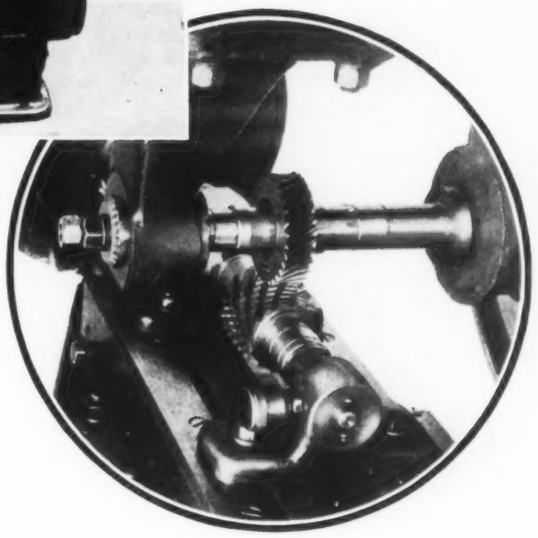


Fig. 3—Cutting a one-to-one ratio worm gear with a tangential feed on a J. E. Reinecker universal hobber

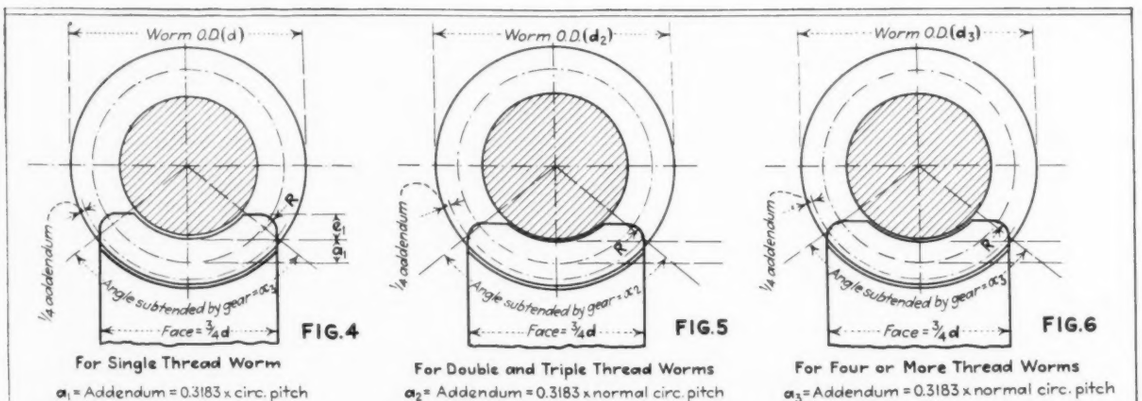
Standard Pitch and Minimum Teeth

General practice requires that only standard circular pitches be used for worm gearing, rather than standard diametral pitches so popular for spur gearing. The tooth proportions, such as addenda and dedenda, are based on the normal circular pitch, except for single threaded worms wherein standard tooth proportions based on the axial circular pitch are permissible. No serious errors are introduced in the latter instance, but difficulties will be encountered with multiple thread worms when their tooth proportions are not modified in some manner similar to the above recommendations.

It is universally recognized that worm gears having 14½-deg. pressure angles and less than 30 teeth are impractical. A 29-tooth gear can be used with a 20-deg. pressure angle and a 27-tooth gear can be employed if the pressure angle is 30 deg. But gears with fewer teeth than the above are not recommended as they would require a certain amount of "dressing up" on the tooth faces. An expedient often adopted for gears of low tooth numbers is to cut the gear "oversize," that is, a certain increment is added to the theoretical pitch diameter of the gear. This new "enlarged" pitch diameter results in the elimination of all interference on the finished gear. This practice cannot be wholly indorsed, especially for multiple thread worms. But it is a useful expedient and can sometimes be resorted to with advantage. In Table I are listed the increments for various numbers of teeth and for gears of 1-in. circular pitch. For other pitches the values of the table must be multiplied by the circular pitch of the gear in question.

Face Width of the Gear

Some of the early practices in determining the face of worm gears are to be deprecated in that the full load carrying possibilities are not realized. The worm gear, unlike other forms of gearing, is a fluid borne friction mechanism and depends largely upon sliding friction to transmit motion. Thus, every worn thread in contact with a gear tooth may be likened to a shaft rotating in its bearing. Therefore, the design should make the contact area between the worm and the gear as


Worm O.D.(d)
R
e1
¼ addendum
Angle subtended by gear = α3
Face = ¾d
FIG.4
For Single Thread Worm
α1 = Addendum = 0.3183 x circ. pitch
R = ¼ circ. pitch
e1 = α1
O.D. of gear = pitch diam. + 4α
Sine of angle α1 = 0.75d1 / d1 − 0.5α1
Worm O.D.(d2)
¼ addendum
R
Angle subtended by gear = α2
Face = ¾d
FIG.5
For Double and Triple Thread Worms
α2 = Addendum = 0.3183 x normal circ. pitch
R = ¼ circ. pitch
e2 = 0.75α2
O.D. of gear = pitch diam. + 3.5α2
Sine of angle α2 = 0.75d2 / d2 − 0.5α2
Worm O.D.(d3)
¼ addendum
R
Angle subtended by gear = α3
Face = ¾d
FIG.6
For Four or More Thread Worms
α3 = Addendum = 0.3183 x normal circ. pitch
R = ¼ circ. pitch
e3 = 0.5α3
O.D. of gear = pitch diam. + 3α3
Sine of angle α3 = 0.75d3 / d3 + 0.5α3

Figs. 4, 5 and 6—Approved method for dimensioning worm gears, formulas for proportioning the teeth and method for determining the central angle subtended by th gear tooth

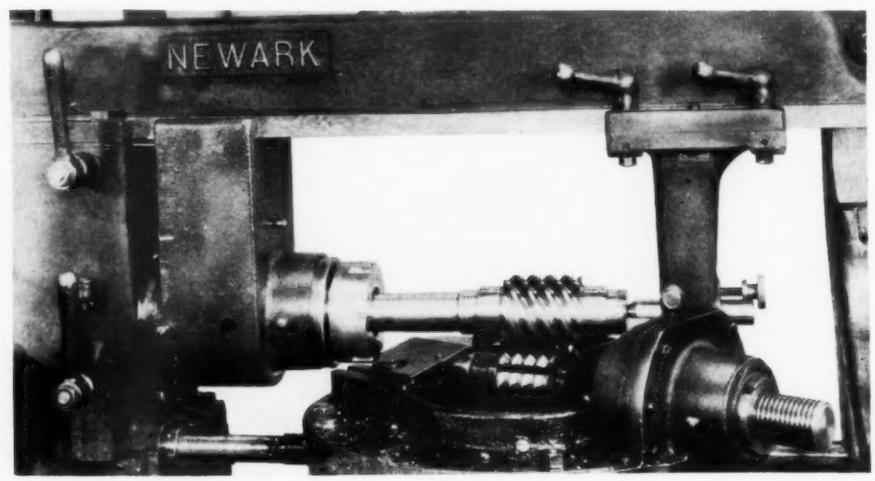


Fig. 7—Worm hobbing, the latest development in the production of highly accurate cut worms, as done on a hobber built by the Newark Gear Cutting Machine Company

large as possible. The writer's practice has been to make the face of the gear 75 per cent of the outside diameter of the worm, giving a subtended angle on the worm cross-section of between 107 deg. and 118 deg. This simple rule gives results comparable with those obtained from the more complicated formulas.

The approved dimensioning of worm gears is shown in Figs. 4, 5 and 6, for worm gears meshing with single, double and triple thread worms, and worms having four or more threads. These figures also give the formulas for tooth proportions and the method for determining approximately the central angle subtended on the worm by the gear face. This last is made use of in determining the maximum load that can be imposed on the gear tooth. As each of the constructions illustrated in these three figures is for the same worm pitch diameter, the same axial pitch and the same number of teeth in the gear, the illustration gives visible evidence of the effect of the number of threads in the worm upon the tooth proportions.

Determining Worm Pitch Diameter

For worms keyed to their shafts, the pitch diameter of the worm can be determined by the formula:

$$\text{Worm pitch diameter} = 2.4\ \text{C.P.} + 1.1\ \text{in.}$$

In this equation C. P. designates circular pitch. For such worms we have:

$$\text{Maximum bore} = \text{C.P.} + 0.625\ \text{in.}$$

The above equations result in worms with a relatively small helix angle and consequently a low efficiency. It is much better to design the worm with as small a root diameter as possible, consistent with its resistance to bending, and thereby obtain a larger helix angle and greater efficiency. Such minimum diameter worms must naturally be integral with their shafts. For an empirical proportion, the following suggested minimum pitch diameters have been found to work out well:

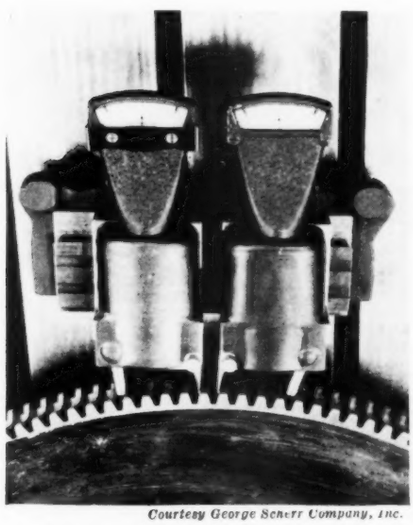
Minimum Worm Diameter Per Inch Axial Pitch

NUMBER OF WORM THREADS	WORM PITCH DIAMETER
Single	2.795 in. to 2.502 in.
Double	2.592 in. to 2.418 in.
Triple	2.502 in. to 2.339 in.
Four or more	2.418 in. to 2.265 in.

There may be instances wherein the above recommended proportions cannot be followed exactly, but usually they will be found highly satisfactory. However, regardless of what rule or formula may be followed to determine the worm pitch diameter, the root

diameter should always be sufficiently large so that the worm will be stiff enough to prevent it from springing away from the gear because of the bending moments imposed upon it.

Because the reactions and bending moments are troublesome to calculate, Table II is given here to make such arithmetical work unnecessary. For the three pressure angles commonly used, the table gives the bending moment load BLw in terms of the end thrust ETw on the worm for helix angles ranging from $1\frac{1}{2}$ deg. to 45 deg. The worm end thrust given in the table is, of course, exactly equal to the turning force on the gear and need only be multiplied by the pitch radius of the worm gear to determine the gear shaft torque. This table also gives the values of the worm torque Tw in in.-lb. at the worm pitch line, the normal tooth pressure NTP and the gear bearing load BLg. All of these are given in terms of the worm end thrust. If any one of them is determined for any worm and gear set, all of the other factors can be found readily by simple calculations. For helix angles that are not included in the table, the values can be interpolated without introducing serious errors.



Courtesy George Scherr Company, Inc.

Fig. 8—Both the worm thread and the gear tooth must be checked for accuracy in order to assure the perfection necessary for high efficiency and durability

Length of Worm Thread

Measured axially, the length of the thread portion of the worm should vary with the number of teeth in the worm gear. A convenient empirical formula is:

$$\text{Length of worm} = \text{C.P.}\ (4.5 + N/50)$$

In this formula N is the number of teeth in the gear. The formula will be found to give excellent results for worm threads that are not ground. But where thread grinding is required the face length of the worm must be increased so that the grinding wheel chatter marks at the end of the thread or threads will not come in contact with the gear teeth. To allow for grinding, the face length of the worm as obtained from the above formula should be increased by 1 in. for circular pitches up to $1\frac{1}{4}$ in., and by $1\frac{1}{2}$ in. for pitches from $1\frac{5}{16}$ in. to 2 in. For pitches greater than 2 in., the length of the worm face should be increased by 2 in.

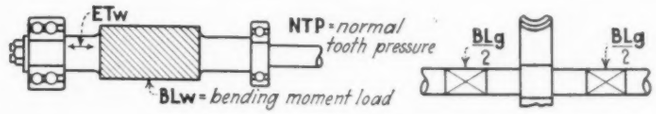
Manufacturing Consideration

An extremely important item is the quality of the workmanship in the manufacture of the worms and gears and their assembly. For heavy-duty drives or high-speed applications, or both, the highest attainable accuracy in the manufacture of both the worm and gear is of paramount importance. It goes without saying that accurate cutting and grinding of the worm must go hand in hand with accurate hobbing of the worm wheel. Without the one, the other is an economic waste and unless both be attained it would be folly to use the highest grade of worm gear materials. A noisy worm gear drive can always be traced to inaccurate manufacture of the units or faulty assembling.

Assuming that adequate and rigid supports for both worm and gear are provided, and that the worm and gear have been made to the required degree of accuracy and finish, correct functioning of the drive will depend upon its proper assembly. It is highly important that there be a full tooth bearing between the worm and the gear. When this is attained the efficiency will be maximum, operation will be quiet and a long life will be assured. Deficiency in any one or all of these items will result in noisy gears, inefficiency with consequent over-heating, and an early failure of the drive. Even inaccurately cut gears when carefully assembled and rigidly mounted will give a fairly good account of themselves in service, whereas the most carefully cut gears, improperly mounted, will wear out rapidly.

TABLE II

Bending moment load, worm torque, normal tooth pressure and gear bearing load in terms of the worm end thrust



HELIX ANGLE (β)	Worm End Thrust ETw	Worm Torque Tw	PRESSURE ANGLE (ϕ) 14½ Deg.			20 Deg.			30 Deg.		
			NTP	BLw	BLg	NTP	BLw	BLg	NTP	BLw	BLg
1½ deg.	1.0	0.026	1.030	0.259	1.030	1.048	0.364	1.048	1.150	0.579	1.149
2 deg.	1.0	0.035	1.031	0.260	1.031	1.049	0.365	1.049	1.158	0.581	1.150
2½ deg.	1.0	0.044	1.032	0.262	1.032	1.050	0.366	1.050	1.163	0.582	1.150
3 deg.	1.0	0.052	1.033	0.264	1.033	1.051	0.367	1.051	1.166	0.583	1.150
4 deg.	1.0	0.070	1.035	0.266	1.035	1.052	0.371	1.052	1.170	0.585	1.151
5 deg.	1.0	0.088	1.038	0.275	1.038	1.053	0.375	1.053	1.177	0.588	1.151
6 deg.	1.0	0.105	1.040	0.284	1.040	1.056	0.380	1.054	1.185	0.590	1.152
8 deg.	1.0	0.141	1.042	0.296	1.042	1.059	0.394	1.056	1.190	0.593	1.152
10 deg.	1.0	0.175	1.052	0.315	1.046	1.063	0.404	1.058	1.195	0.597	1.153
12½ deg.	1.0	0.222	1.066	0.333	1.055	1.075	0.422	1.060	1.200	0.612	1.153
15 deg.	1.0	0.270	1.080	0.378	1.070	1.084	0.445	1.062	1.220	0.700	1.154
17½ deg.	1.0	0.312	—	—	—	1.092	0.468	1.064	1.239	0.733	1.154
20 deg.	1.0	0.357	—	—	—	1.109	0.500	1.066	1.265	0.762	1.155
25 deg.	1.0	0.475	—	—	—	—	—	—	1.285	0.788	1.155
30 deg.	1.0	0.588	—	—	—	—	—	—	1.355	0.824	1.156
35 deg.	1.0	0.714	—	—	—	—	—	—	1.430	0.930	1.157
40 deg.	1.0	0.833	—	—	—	—	—	—	1.500	1.000	1.158
45 deg.	1.0	1.000	—	—	—	—	—	—	1.600	1.160	1.160

PRODUCT DEVELOPMENTS

Front-Wheel Suspension for 1934 Automobiles

In tune with the times, automotive engineers have made radical changes in their designs. Never before in automobile history have the new models been so drastically different with respect to the engineering principles used in their design. It is true that other years have witnessed startling innovations. But this year the changes are basic, in some instances involving a complete departure from the hitherto conventional practices in automobile design. Undoubtedly many of the new constructions that have been incorporated in the 1934 automobiles will, in principle, find applications in other fields.

Although not the only new feature, independent wheel suspension is attracting greatest attention. Nor is this feature entirely new, European cars having used independent wheel mountings for a number of years. But in the heavier and more powerful American cars the problem has been quite different.

General Motors has announced that its whole line of cars will have independent front wheel suspensions with helical springs. Two general types of designs will be used, the one described here will be applied to the heavier cars, Buick, Cadillac, LaSalle and Oldsmobile. Another type of design, details of which are not as yet released for publication, will be used on the Pontiac and Chevrolet cars.

According to F. A. Bowers, chief engineer of Buick, and W. J. Davidson, engineer of General Motors, in the 1933 Buick cars the front spring rate was approximately 300 lb. per in. and the rear springs 150 lb. per in. One object was to reduce the front spring rate to about that of the rear spring, thereby increasing the riding comfort. But leaf springs having the desired reduced spring rate could not be designed satisfactorily because of stress and space limitations. To attain the desired reduced spring rate would have required excessively long springs, and the unsprung weight would have been increased, both contributing to decreased riding comfort. Cost was another factor. The abandonment of the conventional leaf spring was the only solution.

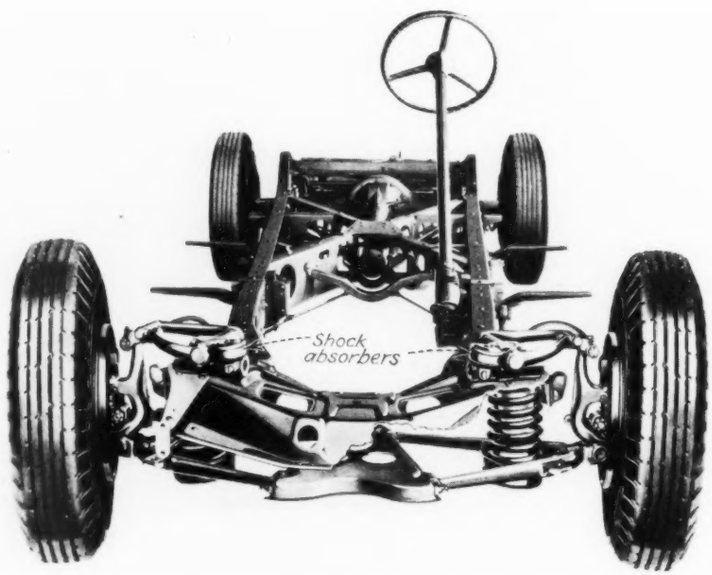
The problem was to devise a type of front wheel suspension that would permit a design wherein the front spring rate could be reduced as desired and still give the rigid support required for proper steering. After considering several possible types of independent front wheel suspension, the one shown in the accompanying illustrations was finally adopted. Its principle of operation is based on the parallelogram linkage. The wheels are mounted directly to the frame through two "wish bones," one above the other and hinged to the frame. The manner in which they support the wheels can be seen readily from the illustration. The springs do nothing but carry the load and are not subjected to any driving or braking stresses. The wheels are free to move up and down but cannot move in any other direction. The springs allow 4 in. compression and 4 in. rebound.

To take care of manufacturing variations, an adjustment is provided so that the wheels can be set to the correct caster angle. This adjustment is made at the time of assembly and need not be touched thereafter.

Camber can be changed by shimming at either end of the steering knuckle support. No shims are used in the original assembly as the proper camber is obtained by correct machining.

A shock absorber is provided for the upper arm, which is somewhat shorter than the lower one. The two arms have been proportioned so that as the whole wheel suspension moves up and down it pivots about the point where the tire tread makes contact with the ground. Therefore there is no sidewise rubbing of the tire on the road as the wheels move up and down.

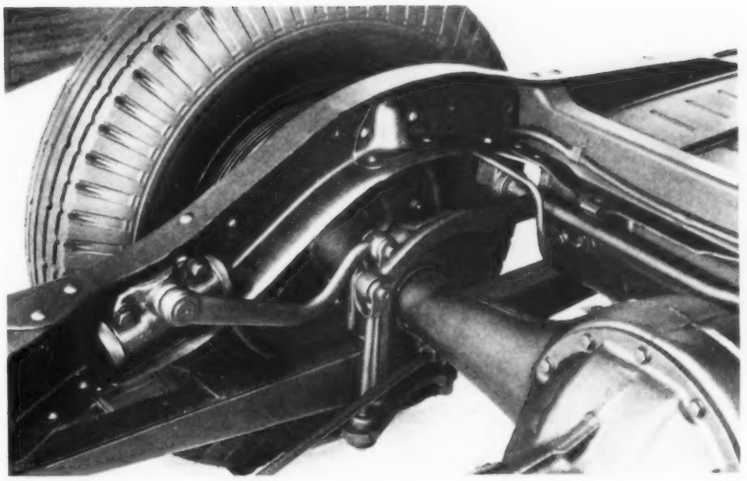
In this new suspension the frame must carry loads formerly carried by the front axle. The front end had to be made more rigid, being



Showing the design of the independent front wheel suspension as developed for the 1934 Buick cars

reinforced by braces between the X-bars and the frame side members. Under the center of the front frame cross member is riveted a heavy triangular plate. The lower wheel support arms are hinged to this piece on either side. This provides a single piece that can be made and assembled accurately to the frame, thereby assuring proper alignment of the wheels.

Because the primary purpose of the independent wheel suspension is to make possible a decreased spring rate, and Buick engineers were not aiming to decrease the rear spring rate which was considered sufficiently low, the conventional leaf springs are used for the rear axle. The torque rod construction at the rear, adopted by Buick several years ago, relieves the rear springs of braking torque and driving stresses.



Leaf springs are still being used for the Buick rear suspension. The torque rod shown here tends to equalize the spring action on both rear wheels and relieves the rear springs of braking torque and driving stresses

Breaking of a front coil spring does not interfere with steering or braking. To eliminate surface flaws that might give birth to fatigue cracks the spring wire is centerless ground before being coiled. As to relative weights, the coil spring is less than one-half the weight of an equivalent leaf spring.

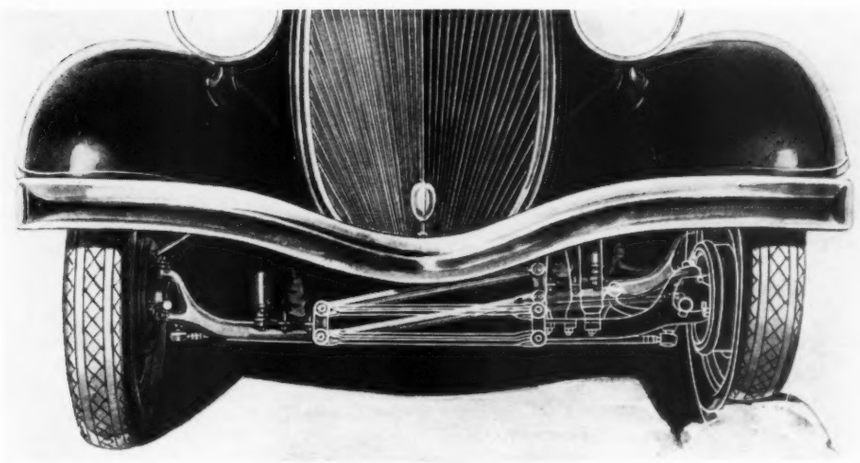
In the accompanying picture is shown the steering gear for the 1934 Buick. The old tie rod has been eliminated. Ball joints at the drag link and steering knuckle arm permit the wheels to move vertically.

Buick's independent front wheel suspension includes the design of an entirely new steering mechanism. Roller bearings are used throughout

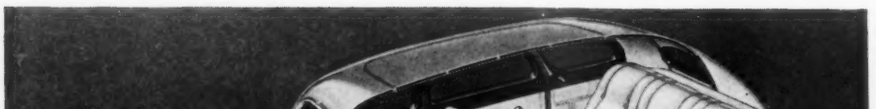
Aiming at the same objective, Hudson engineers developed what they call the "Axleflex" method of front wheel suspension. This design uses the same parallelogram principle as that used by General Motors although the details are different. The independent wheel action, confined substantially to a vertical movement, is also obtained through a parallelogram mounting as shown in the illustration.

The parallelogram construction consists of two rigid drop-forged links pinned at both ends and mounted one above the other. These links have integrally forged forks at their extremities through which the pins pass. Roller bearings are provided to minimize friction.

In the Hudson design, all the changes are contained in the front axle. No alterations were made in either the steering gear or spring



In the "Axleflex" construction developed for Hudson and Terraplane, the parallelogram linkage operates kinematically the same as the "wish bones" linkage used by Buick. Phantom lines indicate the normal position of the wheels



Cut-away view of the Airflow DeSoto whose features are seats cradled in chromium piping, 745 lb. motor directly over front axle, luggage compartment behind rear seat, more head room, a lower body, reinforced to act as a girder

suspension. The new "Axleflex" assembly can be substituted for the old axle without alteration.

In the accompanying illustration is shown the Airflow DeSoto, whose appearance represents the most advanced ideas in streamlining. The reinforced body serves as a girder.

▸ Horizontal Bending Rolls

An unusual design of wrapping-type bending rolls has been developed by Kane & Roach, Inc. As shown in the illustration, the machine is of the horizontal type. It is used for bending I-beams into rims such as used on cable reels, for which true and uniform circles are required.

The machine has a production of about 45 linear ft. per min. It is driven by a 1,140-r.p.m., 30-hp. constant-speed motor. A forward-reverse-stop push button control station is located convenient to the operator, and in addition the machine is provided with a friction clutch operated by a hand lever which is likewise placed convenient to the operator.

▸ Full-Automatic Protection for Worthington Compressors

Worthington "Feather Valve" motor driven compressors incorporate full-automatic operation and protection. Starting and stopping is merely a matter of pushing a button, even with the compressor fully loaded. Worthington engineers have found low starting torque motors to be entirely satisfactory and adapted to full voltage starting, though special conditions might necessitate reduced voltage starting.

Other electrical control features of this line of compressors include inverse time limit overload relays, undervoltage protection and protection against loss of d.c. field or stalling. For a.c. drive provision is made to re-synchronize the motor automatically when a momentary drop in voltage has thrown it out of step.

The main bearings of these compressors can be provided with temperature relays that open the push button circuit and shut down the machine if an excessive bearing temperature is reached because of oil failure or other cause. Similarly, thermostats can be provided for protection against overheating through failure of the jacket water supply. A mercoid switch actuated by a thermostat shuts down the machine in case of water failure.

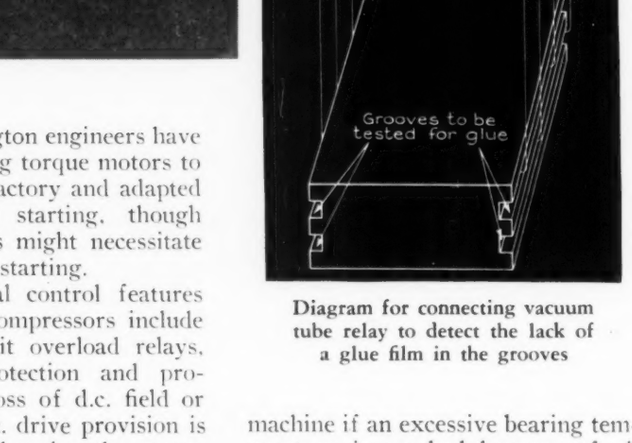
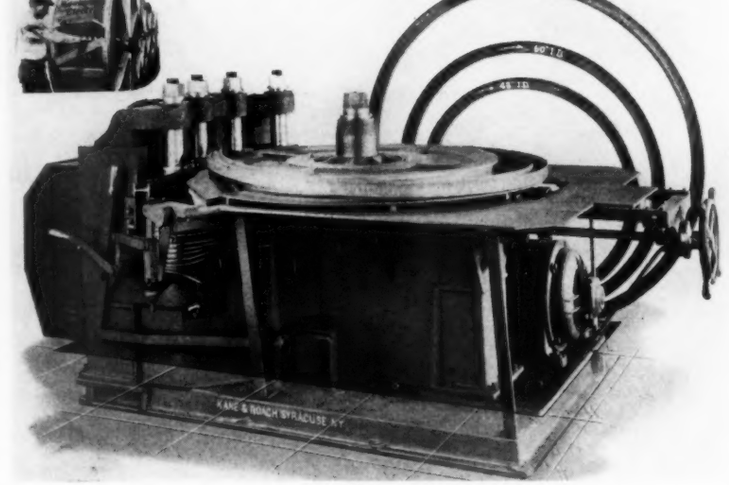


Diagram for connecting vacuum tube relay to detect the lack of a glue film in the grooves



Unusual design of wrapping-type bending rolls developed by Kane & Roach for circling I-beams

▸ Vacuum Tube Relay as a Glue Detector

In the manufacture of box-ends from scrap lumber, certain pieces are tongue-and-grooved, glued and then pressed together. Trouble is sometimes experienced because of the difficulty of detecting the absence of insufficiency of glue between the tongue and the groove. A lack of glue could not be discovered until the box ends pulled apart.

To solve this problem a General Electric vacuum tube relay was connected as shown in the drawing. Steel piano wire contacts scrape in the grooves so that the electric circuit is completed through the glue film. This circuit is connected to the grid of an amplifier tube mounted on a panel, the amplifier tube being connected to control a sensitive relay. If the circuit is open because of an absence of glue, the relay operates to ring a bell and light a lamp.

▸ Portable Loader

Powered by an Allis-Chalmers industrial tractor equipped with semi-steel crawlers, this portable loader developed by the Joy Manufacturing Company is designed for loading any kind of material from ground storage into trucks or cars. It has been particularly successful in loading sugar beets, maintaining a uniform flow of beets. This is largely accounted for by the design of the gathering arms whose oscillation furnishes a steady supply of material to the cleated belt conveyor.

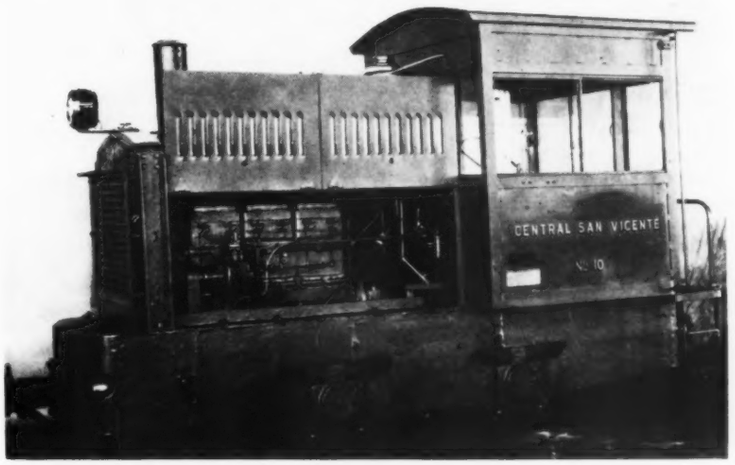
The specially designed power take-off is located between the tractor engine and tractor clutch. This makes it possible to drive the conveyor and gathering arms continuously from the engine and to use the traction intermittently as needed for feeding the machine into the pile of material.

▸ Chain Drive Locomotive

For service in Puerto Rico, the Whitcomb Locomotive Company designed and built this 12-ton 0-4-0 diesel locomotive with mechanical drive. It will haul 324 trailing tons at 10.82 mi. per hr. and 190 tons at 16.87 mi. per hr., thus affording a wide range of effectiveness for industrial haulage.

The locomotive frame is a unit casting with integral engine and transmission mountings, thus insuring accurate and permanent alignment. Semi-elliptical springs are used, with full cross equalization, which enables the locomotive to negotiate rough tracks without derailment.

Power is furnished by a 6-cylinder, $4\frac{7}{8}$x6-in. full diesel engine, rated at 85 net hp. at 1,200 r.p.m. The clutch, designed and manufactured by the locomotive builder especially for locomotive service, is of the multiple disk, cork insert type running in oil. Eight pairs of $15\frac{1}{4}$-in. diam. disks are used.



Four-speed selective transmission with roller chain drives to the axles is used in this 12-ton 0-4-0 diesel locomotive built by Whitcomb Locomotive Company

Transmission is by selective gears, with four speeds forward and four speeds reverse. Drive is from a jack shaft directly to the rear axle by two roller chains and from rear axle to front axle by one roller chain. No guards or housings are provided for the chains. Only two gear reductions are employed, thus delivering a high percentage of the engine power to the driving wheels.

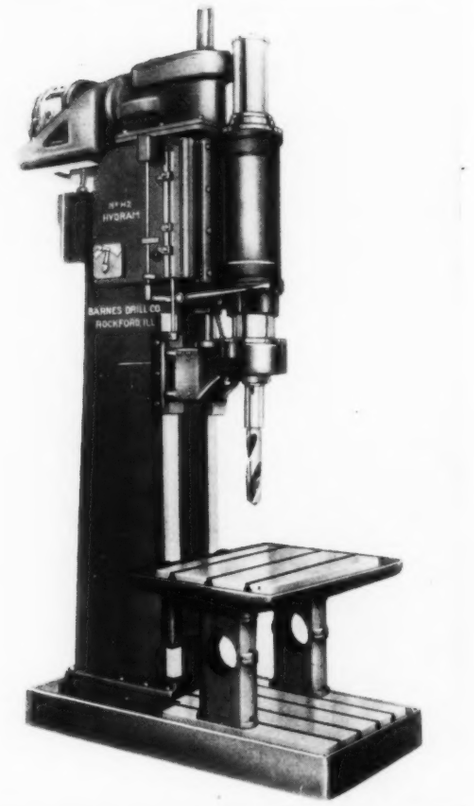
Special attention has been given to providing a clean cut design with a maximum of accessibility and convenience for the operating and maintenance personnel as evidenced by the accompanying illustration.

The Whitcomb series of diesel powered locomotives ranges from 8 tons to 30 tons with mechanical drive, and from 30 tons to 60 tons with electric drive. Two engine-generators are used in the larger locomotives.

Strong and durable design is one of the features of this Joy portable loader. The gathering arms at the bottom feed the belt convevor continuously

Hydraulic
VERSUS
Mechanical Feed–II

A. L. DeLEEUW
Consulting Engineer, Plainfield, N. J.



In this concluding article the author emphasizes the conditions or requirements under which either mechanical feed or hydraulic feed may be expected to be preferable as determined by their relative inherent advantages and shortcomings

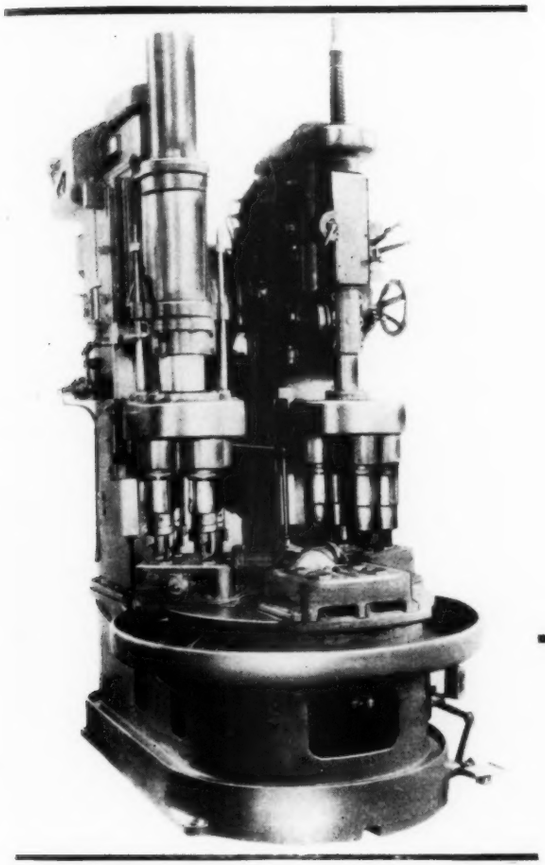
AS TO whether the hydraulic or the mechanical feed calls for a more complicated mechanism, this is a point which it is hard to decide in a general way. It all depends. For example, when a cycle of operations calls for a long stroke, a fast approach, an interrupted feed and a fast return, it would be difficult to imagine a mechanical arrangement that would be as economical to manufacture as a hydraulic one. On the other hand, when the stroke is short, the other requirements being the same, it would be very well possible to use a mechanical feed and even to improve on the hydraulic feed. There are too many possible sets of requirements to talk about this thing in generalities.

That the simplicity of the hydraulic feed—if it is simple—leads to a minimum of maintenance is another thing. It is true that there is not much possibility of a breakage, and looking at breakage alone one does not need to hesitate as to which one of the two systems causes the least amount of trouble. But there are other troubles besides breakages. There is such a thing as leakage; there may be a hesitating or a jumping feed; there may be the necessity of replacing packings; there may be failure of a relief valve to act; or air may have crept into the system in some mysterious way. Any one of these troubles is easy to correct once it is located, but it is not always easy to locate it. Any good mechanic can repair a feed box when it is out of order, but not every one can find out why some hydraulic feed does not act as it should. Here again, it is not all good with the one and all bad with the other.

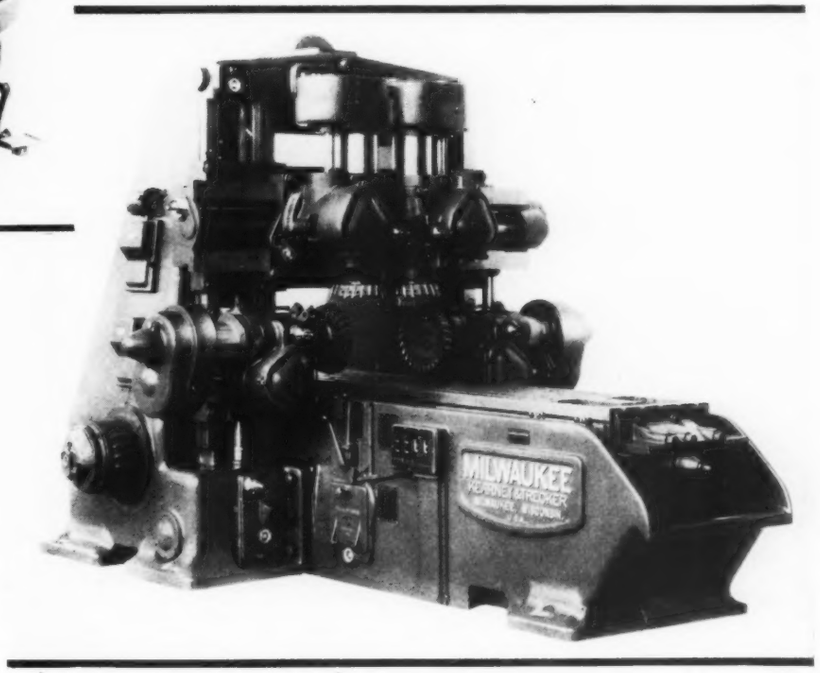
There is still another basis for comparison of the two systems. It is their applicability to various types of machine tools. Of course, we know that the mechanical feed can be applied to any of the existing types, for they are being applied. The question is if the hydraulic feed also can be applied to all these types, and, if so, whether equally good results can be obtained, or perhaps better ones. The claims as to its superiority in some respects have been discussed in the previous paragraphs in a general way. It remains to be seen how its qualities make the hydraulic feed applicable to various types of machine tools and whether these qualities are of advantage or otherwise.

The most clean-cut example is probably that of the drill press. The hydraulic feed lends itself to almost any of the different types of this machine. The radial drill may be called the exception. Hydraulic feed is especially effective when the machine has to go through a complicated automatic cycle. It is true that the same effect can be reached by the use of a cam, but if one wants all the automatic features anl also the possibility of changing the amount of feed by small increments, one is likely to have a complicated piece of cam mechanism. Besides, the hydraulic feed has another advantage, mentioned before, namely, that it prevents the drill from jumping through. To have the full advantage of this latter feature the oil should be controlled at the exhaust end of the pressure cylinder.

Milling machines, boring machines and grinders lend themselves to the application of the hydraulic feed; that is to say, there is nothing in the construction or functioning of these types of machines which makes the use



Left — Hydraulic feed is particularly adapted for drilling. In this Barnes drill the pressure is applied directly over the tool. Above—The left-hand head of this Barnes machine utilizes the hydraulic principle



"Milwaukee" rail-type milling machine designed by Kearney & Trecker. Five individual motors are used for feed and spindle drives

drive, as in screw cutting, then the hydraulic feed is not applicable. It would be possible to use a combination of the hydraulic and the mechanical feed, in which the hydraulic part furnishes the power and the mechanical part the control.

Other applications in which the hydraulic feed cannot be applied without some help, are those machines where more than one element is feeding and where these elements must retain some definite relation. As an example, take the screw machine. Such a machine may have a slide which comes toward the work, first with a rapid motion, then with a slow one, and, perhaps, dwell a moment, after which it retreats rapidly. If there are no other slides working simultaneously the hydraulic feed is quite acceptable; but if there are other slides also, such as cross slides for forming or cut-off, all these slides must work together in a definite way so as to make sure that one does not interfere with another. There is no possible way of maintaining their proper relations if these slides are individually fed by hydraulic cylinders. Some mechanical connection must be provided between the slides and a single hydraulic cylinder must be directly connected to one slide only. Such mechanical connections are, as a rule, undesirable. The reason is that the first slide, that is, the one directly driven by the cylinder, has a slow motion while feeding, and transferring this slow motion to another slide cause its movement to be irregular and halting. Bad work and tool breakage would be the result. When a mechanical feed is used, this difficulty is easily overcome by driving each slide from a relatively fast moving member and maintaining the proper relation between these members. Each slide might be driven by a cam and each cam by a wormwheel, while these wheels are driven by worms which all derive their motion from one common shaft.

On the other hand, the hydraulic feed has some

of this type of feed impossible or inadvisable. It depends on the individual design whether the hydraulic feed will be better than the mechanical feed.

Machines requiring an intermittent feed, such as planers and shapers, do not lend themselves well to the application of the hydraulic feed, though it would not be impossible to arrange them for it.

When applying the hydraulic feed to long machines, it becomes of the greatest importance that the piston rod of the hydraulic cylinder be in tension in whatever direction the feeding element may go. This would apply to grinders with long beds and to lathes. However, there is something else to take in consideration in the case of lathes. If the machine requires that there shall be a definite relation between the feed and the

advantages over a cam feed. The path of a cam, where it changes from a forward to a reverse motion, must have a curve, and this slows the feed before the end of the stroke; whereas a hydraulic cylinder can give an even motion through the entire stroke.

There is another aspect to this matter. It is desirable to slow down the feed when facing has to be done at the end of the stroke of the main slide. This can be accomplished in a cam feed by reducing the slope of the cam groove. A flat part may come after this reduced portion, so that there is a dwell. In this manner one obtains an almost constant feed during the turning or drilling part of the stroke, a reduced feed while facing and a dwell to clean up the face and, at the same time, get a greater accuracy as to the length of the piece of work. There is no delay in the start of the return stroke, for the dogs may be set so that this return stroke starts while the roller is still on the flat part of the cam.

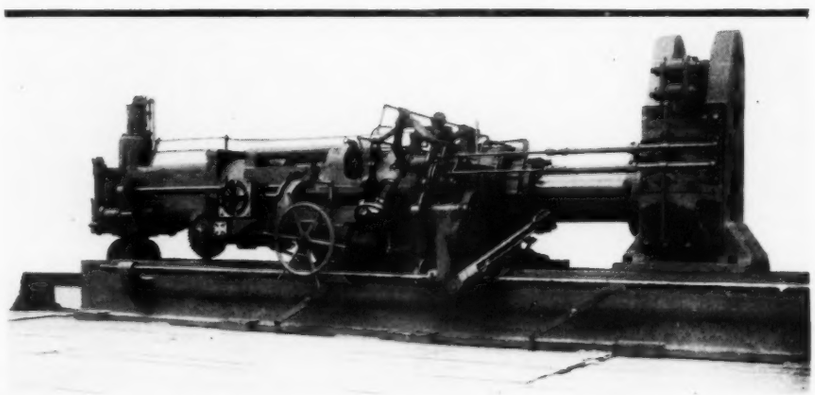
Though the hydraulic feed does not lend itself readily to a reduction of the feed while facing, it is, on the other hand, specially adapted to give a dwell. Moreover, it is possible to let the slide come against a positive stop without danger of breaking something or stalling the machine. The same thing is possible with a mechanical feed, but at the expense of some complication. It requires a slide in the slide, the first one bearing against the second through the intervention of a loaded spring. This spring should be loaded sufficiently so that it is not further compressed by whatever pressure is needed for the cut. When the slide has reached the stop the spring allows some further movement of the auxiliary slide. No controlling devices are needed.

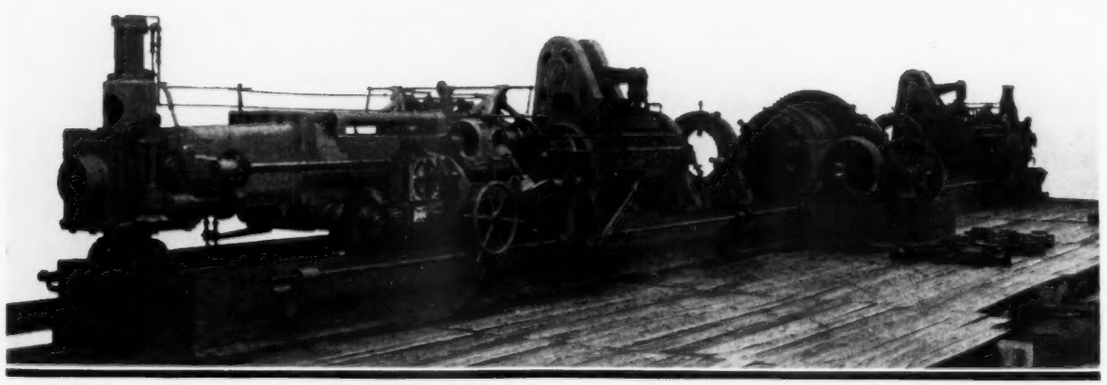
However, such controlling devices are needed when hydraulic pressure brings the slide against a stop. In the mechanical arrangement there was still some moving part to operate on a dog, though the slide itself had come to a standstill, but this is not the condition when it is moved by a hydraulic cylinder. Some means must be found to move a valve when oil no longer flows into the cylinder. Various means may be employed for this purpose, but whatever means are used, something is added to the complication of the feed, which may be a decided disadvantage.

Generally speaking, a cam will do the same as a hydraulic feed for machines which have to go through a definite cycle of movements, provided the stroke is not too long. This limitation as to the length of stroke does not exist when hydraulic feed is used. In theory a cam can be made of any size, but in practice this is not so, unless one wants to forget the economical aspect of the design.

In a machine the writer designed in 1903 hydraulic feed was used with a stroke of thirty feet, and this was not at all the limit of what could have been done. As in an earlier machine of shorter stroke, the rate of feed was controlled by mechanical means. Such a combination has many possibilities which by no means have as yet been exhausted.

Finally, the writer wishes to call attention to the use one can make of hydraulic pressure for the purpose of indexing. To give an example, the writer had occasion to design a machine in which there was a turret of a diameter of seven feet and weighing many tons with the various tools and devices on it. Of course, the ordinary means for indexing could not be used, and refuge was taken to hydraulic indexing, which was rather simple in design and positive and quick in its action.



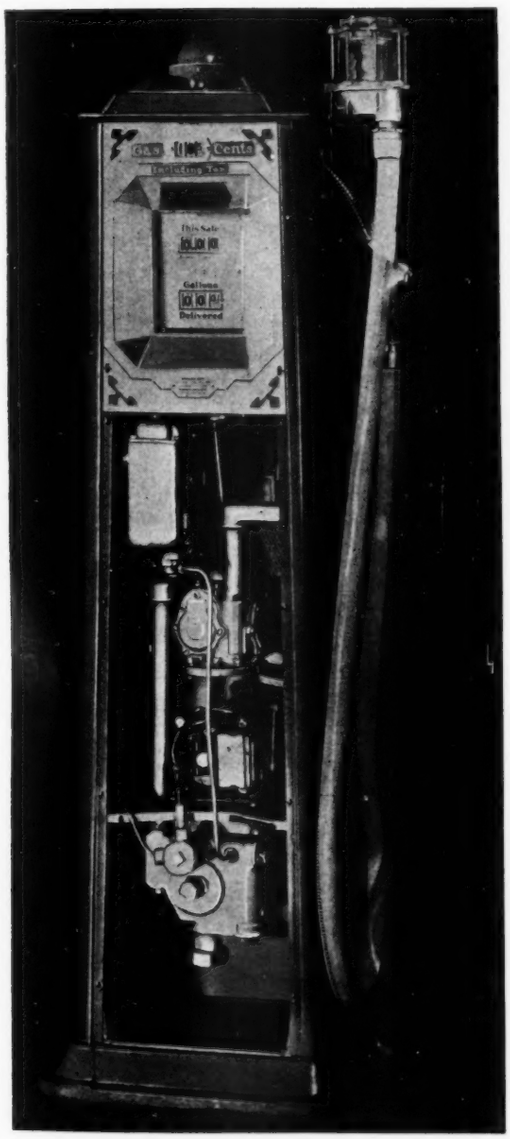


Hydraulic feeds are not new. In 1904, Mr. DeLeeuw designed this hydraulic boring machine built by the Niles Tool Works Company for Midvale Steel. The machine was designed for boring holes 6-in. to 14-in. diameter in forgings up to 60-ft. long and 23-in. outside diameter. Upper picture is a close-up view of the boring head

Case Histories In Product Design—VII

T. J. MALONEY
New Jersey Zinc Company

Brief descriptions of successful products whose designs were based on a correct interpretation of the relative values that the consumer attached to utility, beauty, convenience and durability



In this Wayne pump a register is set to the price per gallon. Any number of gallons or the quantity of gasoline costing any specified sum can then be delivered to the customer

BECAUSE it is interesting and valuable to know the underlying reason for the success of products that have met consumers' needs, this second series of Case Histories will delve deeper into the underlying reasons. By the same token the data on design and manufacturing will be enlarged in order to give a more complete picture. Even with the necessarily abbreviated description that will be given each product, there will be a "follow through" from the conception of a new idea to its ultimate sales success, thereby giving a greater insight into the whole problem of planning, making, selling and consumer acceptance, as it applies to each of the products described.

In this first instalment we present what is undoubtedly the most striking new development in merchandising equipment, the new Wayne gasoline pump. And turning to the field of home equipment, we describe the new Apex vacuum cleaner, a design that emphasizes operating features that are expected to appeal to the housewife, and the new General Electric fan.

The Wayne Gasoline Pump

Case History No. 19

Manufactured by the Wayne Company, this product is a result of the combined efforts of the engineering division of that company and engineers of the Veeder-Root Company who designed the registering mechanism.

Design Aims—In the act of purchasing gasoline, perhaps the most aggravating thing for the customer is to be required to wait for change and then to put the change into the pocket or pocketbook. If the customer could hand the station attendant the exact amount of

money for his purchase, he or she could immediately start off again on the journey. This would be possible with a dispensing unit that could be set to deliver automatically the exact quantity of gasoline costing any "even amount" of money that might be tendered by the motorist. The motorist would then be saved the time and annoyance involved in making change. This would serve to establish popularity of the service station with the customer and, in addition to this, the owner of the

A twist of the pistol grip on the handle steers the Apex vacuum cleaner, a headlight shows up the floor dirt

service station could be fairly certain, at least at present prices of gasoline, of getting a greater volume of sales. With gasoline prices usually ranging between 15 and 20 cents, many of the 5 and 10 gallon orders would be changed to one or two dollar orders, thus assuring an increase in sales.

Design and Manufacture—With the design aim established, the Wayne Company sought the assistance of the Veeder-Root Company to develop a simple and foolproof mechanism to register accurately and simultaneously the price and amount of gasoline delivered by the pump. With the register there had to be combined an easy method of adjusting the recording mechanisms to take care of the price fluctuations.

Veeder-Root Company solved the registering problem with a mechanism that has met all requirements. A full description of this device would be an article itself. Most of the intricate gearing and working parts are zinc die-castings. As to the accuracy and reliability of the registering mechanism, more than 4,000 pumps have been sold and put in service without a single complaint.

The accompanying photograph of the finished Wayne pump shows the three recording dials. These show the price of the gasoline per gallon, the total price for the sale, and the number of gallons delivered on that sale. The last two registering dials operate synchronously and can be set for any price per gallon.

Cast and wrought iron parts, forgings, zinc die-castings, steel and copper tubing, are the principal materials used. The cast iron pump base is joined to the top casting by steel angles. Front and sides are sheet steel. There are registering faces on both the front and the back so that the registers can be read from either side. Both the side and the front sheets can be removed quickly by grasping with a pair of pliers the knuckles that hold the sheets and turning these knuckles through 90 deg. This simplicity speeds up the repair and inspection of the working parts of the pump. As can be seen from the illustration, the pump has a most attractive appearance.

Sales Success—Although the price of this product is approximately 50 per cent higher than the average price of such pumps, the sales quota of 2,500 units for 1933 was doubled. Salesmen's reports declare that this higher selling price was no obstacle when the consumer appeal and merchandising possibilities of the pump were stressed.

Probable Reasons for Success—Behind the clear thinking that foresaw the value of the dual register, was a sound merchandising plan. Sales were all in small units to high-class individual stations. No effort was expended in attempting to sell large orders to the large oil companies with chain outlets. Obsolescence will first do its sales work there.

Apex Electric Vacuum Cleaner

Case History No. 20

Manufactured by the Apex Rotarex Corporation, this product was designed by the engineers of that company with the cooperation of the engineers of the Precision Die Casting Company in the development of the designs of the die-cast parts.

Design Aims—A study of the new Apex deluxe electric vacuum cleaner seems to indicate that the engineering division and sales staff of this company concluded that engineering and operating features whose advantages are readily understood and easy to demonstrate can be made the salient sales points in the appeal to the housewife. Such conclusions would account for the numerous convenience features that are incorporated in this new product.

Design and Manufacture—Practically all the new features of this design relate to the ease and simplicity of operating the machine. The pistol grip on the handle has a control switch convenient to the finger tips. By an easy twist of the handle the user can steer the cleaner, thereby avoiding the necessity of using the base of the machine as a pivot, the handle as a radius, and requiring of the operator countless steps to turn the machine. A headlight is provided to expose dirt and dust hidden in the shadows of the room. A touch of the toe on the conveniently placed nozzle adjuster is all that is necessary to change the distance between the nozzle and the carpet. And in order to be able to gage quickly the height of the position of the nozzle with respect to the floor covering, an easily visible indicator tells at a glance the position of the nozzle. These last two features increase the operating efficiency of the machine.

From an appearance standpoint, this is an extremely attractive machine. It is a homogeneous mass that makes no attempt to conceal its selling points for the sole purpose of heightening its aesthetic appeal.

For lightness, aluminum die-casting (five in number) are used. For those parts where engineering strength is required, zinc die-castings, fifteen in number, are used. The handle is of stainless steel tubing. A compressed rubber gasket between the upper and lower nozzle cases is used for sealing the joint and also damping out the vibrations. The two-speed motor is rubber mounted. For finish, the handle is nickel plated, the body and motor housing polished aluminum.

The necessity of a fine surface over the entire area of the body casting means that inspection is severe and rejections high. Would it be better to break up that surface by introducing a decorative motif at the expense of its present simplicity, thereby cutting down rejections? Or can something be done in the casting process or die making to insure a greater uniformity of castings? This problem has not yet been solved.

Sales Results—Placed on the market in October, the orders exceeded expectation. The machine was so well received that top production lagged behind the order taking of the sales division, and in the middle of December the plant was still behind in filling outstanding back orders.

Probable Reasons for Success—In many household devices the selling tendencies seem to be to call attention to the ensemble and then elucidate the technical and gadget advantages. But this company has chosen the procedure of selling the superiority of this $49.50 model by concentrating on its technical features. And these features are incorporated in the machine in such a way as to be as obvious and distinct to the consumer as they are assets to the machine. And this company has an excellent claim for the effectiveness of its sales procedure. In fact it has the best of all claims, namely, sales success of a quality product in a price market.

General Electric Fan

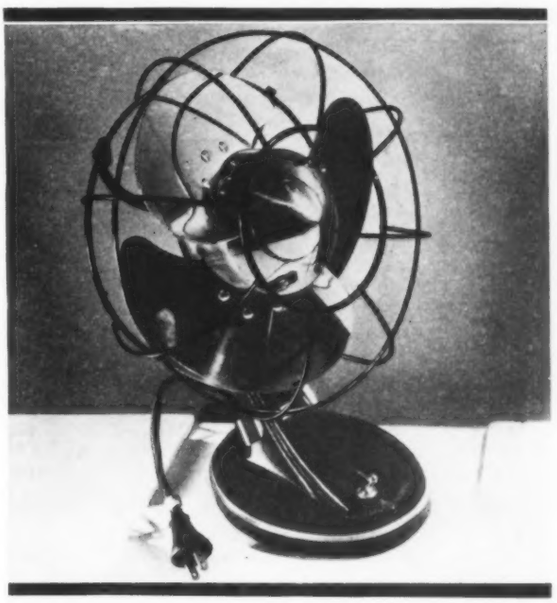
Case History No. 21

Manufactured by the General Electric Company and developed by the engineering department of that company, the new model fan described here is a distinct departure from the conventional design of this line of product.

Design Aims—There are two logical markets for the sale of portable electric fans. One is their use for offices, stores and business establishments. The other possible sales outlet is fans for home use. But as a household article, the electric fan is still considered in the luxury class and housewives will invest in almost every electrical home convenience, except a fan.

A complete redesign of the conventional existing fan was considered necessary. Such a redesign would have to encompass both mechanics and appearance. Silent operation without loss of wind force and no increase in electrical consumption were the most important operating improvements sought. As to appearance, the aim was to make it radically different and more pleasing. And this was to be accomplished with a design whose production cost had to be within the present price range for medium size electric fans.

Design and Manufacture—The accompanying picture shows the finished result of this redesign. The conventional four-blade fan has not only lost a blade, but has been given a much deeper pitch. These changes in design make the fan perhaps the most quiet ever developed, with an air delivery in volume and velocity equal to that of fans of greater diameter.



General Electric's new fan has three deep-pitched blades riveted to a cup-shaped hub that houses the motor. Arm and base are zinc die-castings

As to construction, the chromium-plated blades are riveted on projecting fins welded to the blunt-nosed projectile-shaped center which also serves to house the fan motor mechanism. The black enameled wire guard has been formed symmetrical with the blade and housing. To increase the stability of the fan, the black enameled arm and base, both zinc die-castings, have a graceful backward sweep, giving the whole assembly a graceful appearance. The small and simple control switch is located at the most prominent and convenient spot on the base. A friction hinge joint between the base and arm permits tilting the fan.

Sales Success—Dealer acceptance was greater than expected and repeat orders quickly doubled production requirements. This in spite of the fact that the fan retailing at $13.95 for a 10-in. blade size made it a relatively high-price quality product when compared to other units of the same size.

Probable Reasons for Success—This product represents a distinct change in portable electric fans. The noiseless feature was a selling point of first importance. But it is the combination of style appeal and fine operating qualities that accounts for the striking sales success of this product.

Typical Designs of

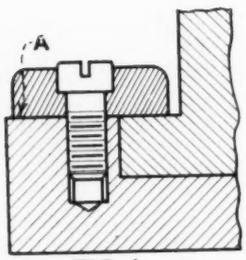


FIG. 1

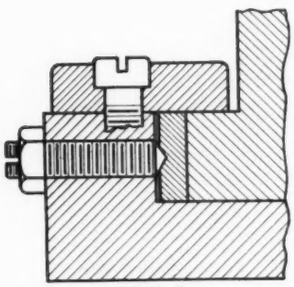


FIG. 2

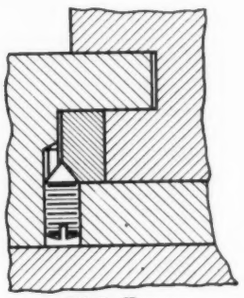


FIG. 3

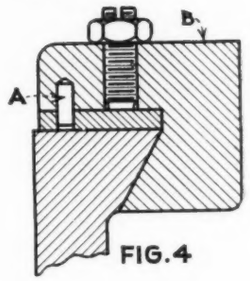
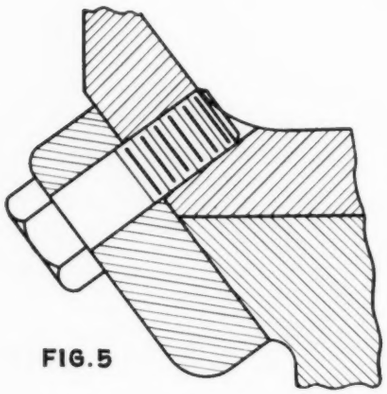


FIG. 4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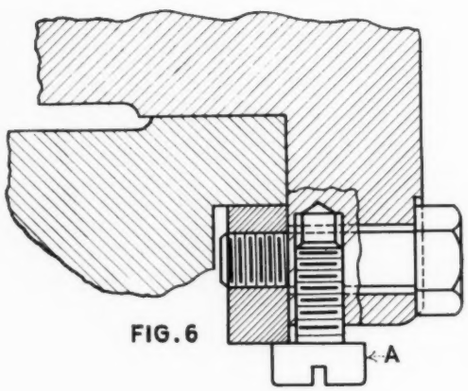

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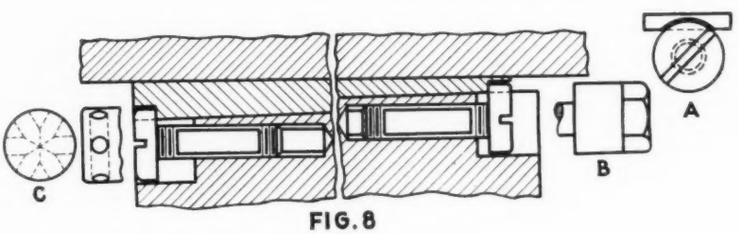


FIG. 8

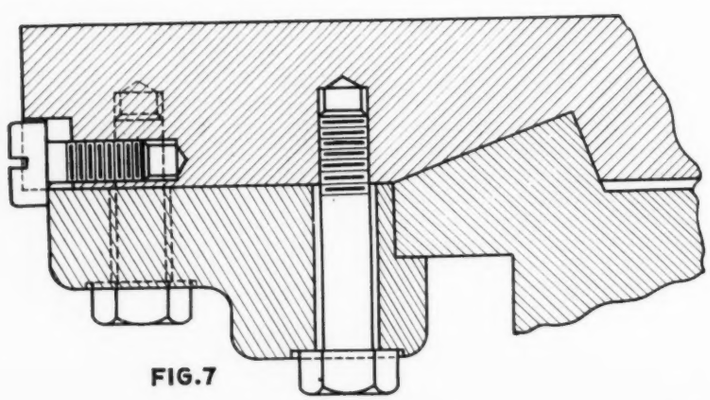


FIG. 7

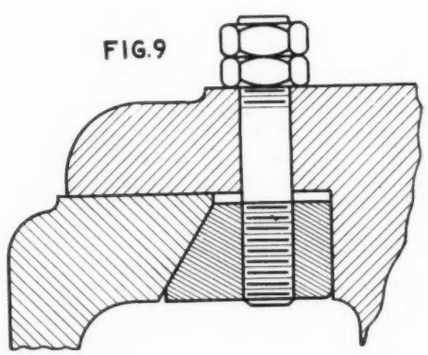


FIG. 9

ADAM FREDERICKS

Fig. 1—Probably the simplest form of guide. It may be machined to use shims of some sort at *A*. When wear takes place, the shim or a part of it, can be removed. Or the guide at *A* may initially be scraped to fit in which case it must be rescraped to compensate for wear.

Fig. 2—A flat gib is used in conjunction with the flat guide as in Fig. 1. It has nut locked screw adjustments at regular intervals. Good only for light loads.

Fig. 3—Another form of adjustment with a cone pointed screw bearing on a bevelled corner of a flat gib. It is necessary that the screws have a snug fit in the threaded holes to keep them from backing out.

Fig. 4—A flat gib used on a dovetail slide. A pin *A* fixes the gib in relation to the sliding member *B*. Flat-end dog pointed screws are used.

Fig. 5—A flat guide on a dovetail. This construction must be sturdy because the gib overhangs the dovetail surface.

Fig. 6—A flat narrow guide adjusted by large headed screws *A* and clamped by hexagon head screws at right angles.

Fig. 7—This design is used in conjunction with the guide shown in Fig. 6 and is adjusted and clamped similarly.

Fig. 8—The flat tapered gib with a slight taper of $\frac{3}{16}$ or $\frac{1}{4}$ in. per ft. is adjusted and locked by opposed screws engaging a counterbored segment in the gib as at *A*. At *B* and *C* are different types of heads for the adjusting screws.

Gibs and Guides—I

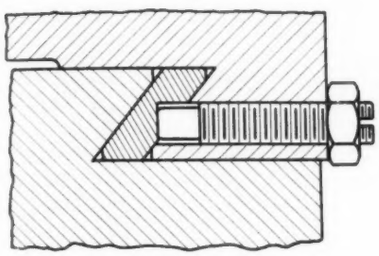


FIG.10

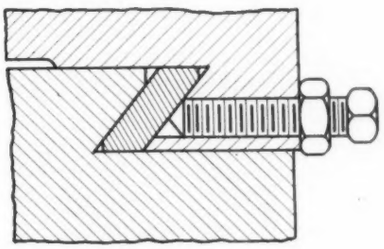


FIG.11

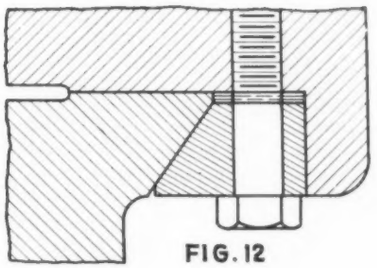


FIG. 12

Fig. 9—This gib has studs fixed in it. Adjustment is obtained with hexagon lock nuts.

Figs. 10 and 11—Simple methods for adjusting flat dovetail gibs. Note that all sharp corners are flattened off with a $\frac{1}{16}$-in. flat.

Fig. 12—This dovetail gib is backed with a flat surface. Shims are used for the wear adjustment. The gib is always clamped solidly.

Fig. 13—While design is similar to that shown in Fig. 12, no shims are used with the gib and it is screw adjusted, the screw being locked in position by check nuts.

Fig. 14—This design is similar to that shown in Fig. 12, but with the difference that it has the addition of a tongue running the entire length of the gib. If no shims are used, the gib must be rescraped to fit when the wear is to be taken up.

Fig.15—By using a dovetail guide such as shown here inserted in the bottom of a table, it is not necessary to have gibs. To compensate for wear some of the shims are removed until the desired fit is obtained.

Fig. 16—A flat tapered gib with a projecting head to facilitate adjustment. The stud is locked in the stationary casting and the gib is adjusted by hexagon nuts on either side of the head. This gib can be used singly or in pairs as shown here.

Fig. 17—Two gibs as in Fig. 16 can be applied to a table Vee way as shown here. Usualy only one gib is needed. But when the length is great it becomes necessary to use two gibs.

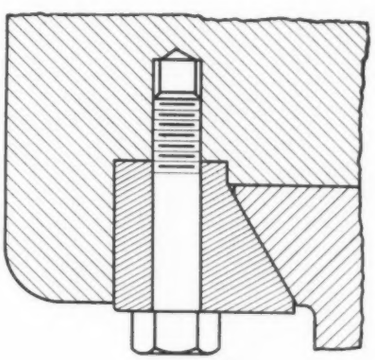


FIG.13

FIG.14

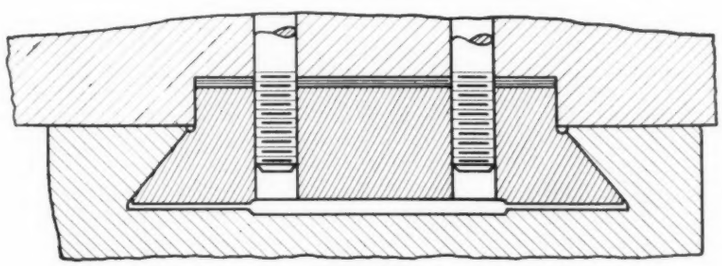


FIG.15

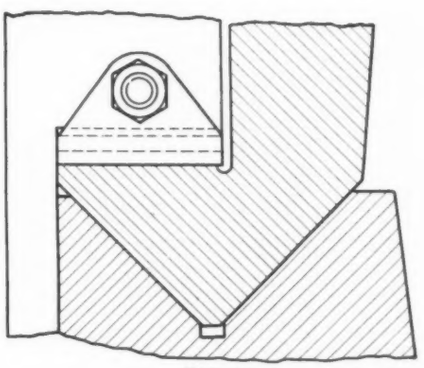
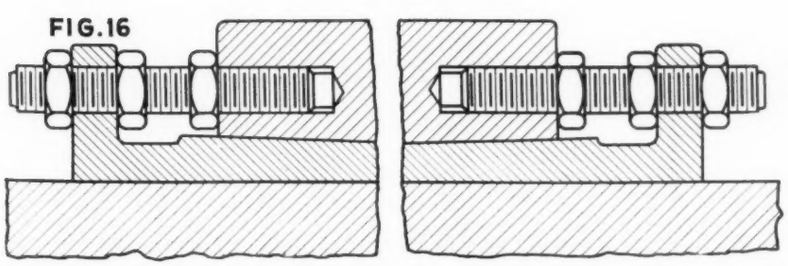


FIG.17

FIG.16



Designing With Laminated Plastics

H. M. RICHARDSON
Plastics Department, General Electric Company

The design of spur gears for strength and wear is given in this article, to be followed by one on the design of bearings and spinning pots

IN THE design of machine elements, the problem of mechanics and kinematics are the same regardless of the material used. Having established the load and the stresses produced, the problem becomes one of balancing the physical property of the material used against the conditions that will be imposed. Experienced engineers who are familiar with the behavior of a given material will base many of the design details upon past experience instead of making calculations. It is this fact which gives rise to the danger of ignoring some important physical property when working with a new material having radically different physical characteristics. The design of typical machine elements made of laminated plastics, as given here, are set forth to emphasize the manner in which the properties peculiar to laminated plastics affect the design of machine parts.

Most failures in mechanical design arise not from a rupture of the material but rather as a result of excessive deflection. Naturally, such failures are most likely to occur when the designer changes from a rigid material such as steel or cast iron to a relatively elastic material such as laminated plastics.

There are two general principles that may be used to avoid failure caused by deflections. Usually, the unit working stress is made low enough so that the amount of deflection will be negligible or harmless. The other method is to design the part and the associated part so that any reasonable amount of deflection will not do any damage.

TABLE I
Variation of Tooth Thickness with Addendum

Addendum 1 d.p.	Tooth Thickness at Pitch Line	
	14½ Deg. P.A.	20 Deg. P.A.
0.00 in.	1.0536 in.	0.8429 in.
0.20 "	1.1570 "	0.9884 "
0.40 "	1.2605 "	1.1340 "
0.60 "	1.3639 "	1.2796 "
0.80 "	1.4674 "	1.4252 "
1.00 "	1.5708 "	1.5708 "
1.20 "	1.6742 "	1.7164 "
1.40 "	1.7777 "	1.8620 "
1.60 "	1.8811 "	2.0076 "
1.80 "	1.9846 "	2.1532 "
2.00 "	2.0880 "	2.2987 "

Gears usually fail by the wearing away of the teeth as the result of one or more of three chief causes. The first is the frictional wear caused by relative sliding between the gear tooth surfaces. This may be minimized by careful cutting to give smooth surfaces and by providing adequate means for lubrication and protection from abrasive substances. The second is the pitting of gear teeth caused by excessive specific compressive stresses at the surface. This may be avoided by careful design, keeping the maximum specific compressive stress well below the compressive yield point of the material. The third is the cutting of the flank of the driving tooth by the tip of the entering driven tooth as the two teeth come into contact.

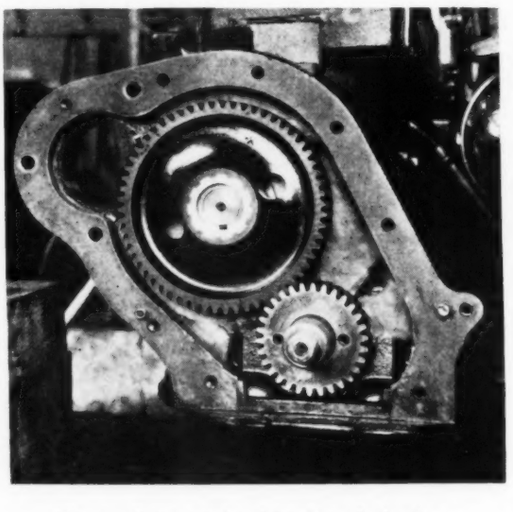


Fig. 1—Gears made of laminated plastics as in an automobile engine timing gear train

In the accompanying Fig. 2 is shown the path taken by the corner of the driven tooth with respect to the driving gear when standard tooth proportions are used. This trochoidal curve is tangent to the flank of the driving pinion and it will be noticed that there is quite a distance along the pinion tooth flank where the tip of the driven tooth almost touches. If the gears could be made with perfect tooth spacing and of a material of infinite rigidity, there would be no interference. But any deflection of the teeth will allow the entering driven

tooth to scrape over the flank of the driver and scuff the surface. This will occur even if the driven tooth has been modified so that its corner is dulled to give less cutting action than would result from a sharp corner. But this does not remove the tendency of interference and it also has the undesirable effect of reducing the duration of tooth contact.

It is evident that regardless of the material used, the proportions of the gear tooth must be modified if the scraping action is to be eliminated. And as the material of the gear becomes more resilient, the importance of taking this into consideration increases. Therefore gears made of non-metallic material of a high resiliency should be designed with modified tooth proportions.

Design of Non-metallic Pinions

Non-metallic drive pinions, such as those made of laminated plastics, owe their quietness of running to the high elasticity of the material. From this it follows that the tooth interference of gears made of such material will be considerably greater than in gears made of steel or cast iron. Therefore, it is desirable if possible to change the design of the teeth so that the trochoidal path of the entering driven tooth along the flank of the driver is shortened or eliminated, thereby materially reducing or entirely avoiding the cutting action. This may be done by increasing the addendum of the driving pinion up to a maximum of all-addendum, and correspondingly reducing the addendum of the driven gear to a minimum of zero. This is illustrated in Figs. 3, 4, and 5. The center distance, pitch circles and base circles remain constant if the same standard cutter is used. The outside diameter of the gear blanks are changed to give the proper dimensions and the cutter or hob is sunk to its usual full depth for cutting. The tooth thickness at the pitch line is increased on the driver and decreased on the driven. For gears with 1 diametral pitch the relation between tooth thickness and addendum for full depth teeth, cut with standard cutters, is:

$$\text{Tooth Thickness} = 1.5708 + 2\,(\text{addendum} - 1)\tan\alpha$$

In the above equation the tooth thickness and addendum are expressed in inches and α is the normal pressure angle. Table I shows the evaluation of the above equation for both 14½-deg. and 20-deg. pressure angles. Values for any other pitch can be obtained by simply dividing by the diametral pitch.

It is not always practical to go to the limit of making the driver all-addendum and the driven all-dedendum because the driver teeth may become too pointed or the driven teeth might have too much undercut. Table II shows the maximum addendum which may be used on the driver and still leave a tooth tip thickness of at least 0.150 in. on a 1 diametral pitch gear. Gears of other pitch will be proportional.

The above discussion applies where the non-metallic gear is the driver and where there are but two gears in a train. Where the non-metallic gear is driven and the driver is steel or cast iron, the scuffing action does not take place and there is little advantage to be gained by the long and short addendum combination except where it is used to avoid excessive undercut on either driver or driven when one of the two has a small number of teeth.

TABLE II

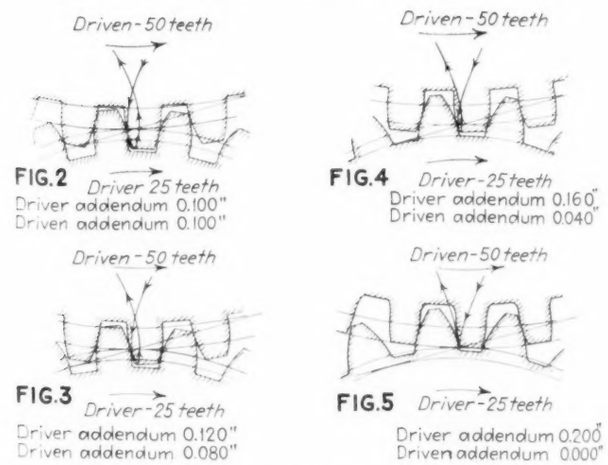
Maximum addendum to give a minimum of 0.150-in. tooth thickness at tip of 1 d.p. tooth

No. of Teeth in Driver	Max. Addendum 1 d.p., 14½ Deg.	Max. Addendum 1 d.p., 20 Deg.
12	1.76 in.	1.67 in.
13	1.81 "	1.70 "
14	1.85 "	1.75 "
15	1.89 "	1.80 "
16	1.93 "	1.84 "
17	1.96 "	1.88 "
18	2.00 "	1.92 "
19	2.00 "	1.97 "
20 or more	2.00 "	2.00 "

With reference to design calculation, every design should be checked to see to it that the duration of tooth contact is satisfactory. This may be calculated by the usual equations such as those given by Earle Buckingham in his book "Spur Gears." From the length of the effective line of action, the number of teeth simultaneously in contact should be calculated. There should be at least 1.4 teeth in simultaneous contact in order to give a smooth running action.

It must be considered that when the gears are made of laminated plastics, the greater resiliency of this material will be favorable to a more uniform distribution of the load and smoother running. Both these factors have a direct bearing on both the allowable static stresses that may be used in the design of the gears and the velocity stress factor that must be applied. Also, the allowable unit working stress will have to be modified when a long addendum drive pinion is used.

With reference to the strength of the gears, the allowable tooth load can be calculated by the Lewis formula. It must be noted, however, that the Lewis formula is based on the strength of a single tooth. It



Figs. 2 to 5—Showing the effect on the trochoidal path of the driven gear tooth with respect to the driving gear tooth when the addendum of the driven tooth is decreased to zero. All the teeth were cut with the same 10 P——14½ deg. standard hob

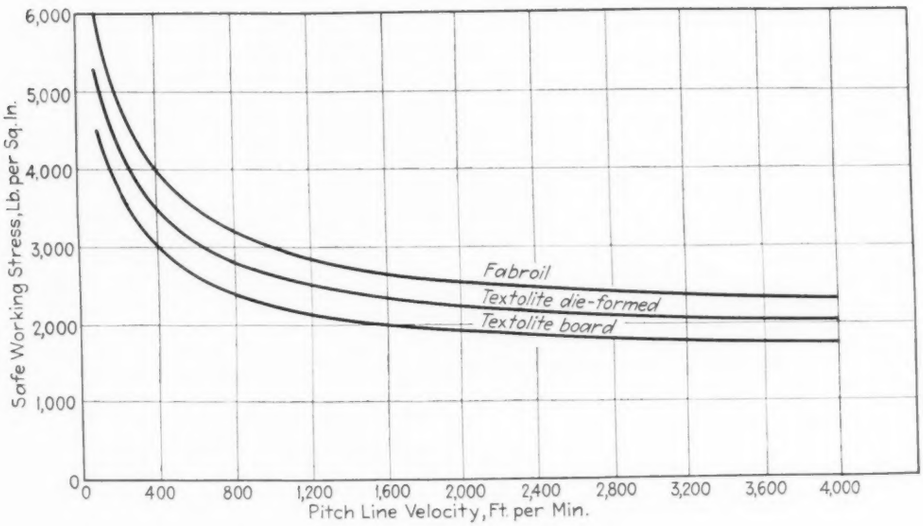


Fig. 6—Values of safe working stresses for use in the Lewis formula for the strength of gear teeth as recommended for three typical laminated plastic materials: Fabroil, Textolite Die-Formed and Textolite Board

presupposes that a single tooth carries the whole load. But with non-metallic gears the low modulus of elasticity of the material permits a tooth deflection under load that is considerably greater than the commercial errors in form and spacing. Therefore, even with commercially cut gears there will usually be two or more teeth sharing the load. When the non-metallic drive pinion has been made with a long addendum to avoid scuffing by the driven gear tooth, the safe load may be

TABLE III
Form Factors for the Lewis Formula

No. of Teeth	Value of Y		
	14½ Deg.	20 Deg.	14½ Deg. All-Addendum
12	0.067	0.078	——
13	0.070	0.083	——
14	0.072	0.088	——
15	0.075	0.092	——
16	0.077	0.094	——
17	0.080	0.096	——
18	0.083	0.098	0.116
19	0.087	0.100	0.117
20	0.090	0.102	0.118
21	0.092	0.104	0.119
23	0.094	0.106	0.120
25	0.097	0.108	0.121
27	0.100	0.111	0.122
30	0.102	0.114	0.122
34	0.104	0.118	0.123
38	0.107	0.122	0.124
43	0.110	0.126	0.124
50	0.112	0.130	0.125
60	0.114	0.134	0.125
75	0.116	0.138	0.125
100	0.118	0.142	0.125
150	0.120	0.146	0.125
300	0.122	0.150	0.125
rack	0.124	0.154	0.124

increased to a maximum of twice that calculated by the Lewis formula. The degree to which this safe load is increased depends on the actual number of teeth sharing the load and the extent to which the long and short addendum combination can be used.

In the accompanying Fig. 6 are given curves showing the safe working stress for gear teeth made of laminated plastics, while in Table III the form factor for the Lewis formula is given.

The maximum specific compressive stress at the contact surface of the gear teeth must be low enough to avoid surface fatigue failure or pitting. Using the Hertz equation for the stress at the surface of two parallel rolls in contact, a formula has been developed to show the maximum load on gear teeth to avoid pitting. This formula, published in the *1931 A.S.M.E. Research Publications*, "Dynamic Loads on Gear Teeth," page 63, is repeated here for convenience:

$$W_w = D\ T\ K Q$$

W_w = limiting static tooth load at the pitch line in lb.

D = Pitch diameter of pinion, in.

T = Width of face, in.

K = Load stress factor depending on pressure angle, fatigue limit of material, and moduli of elasticity of gear and pinion. (For laminated plastics, a tentative value for K is 189 for a 14½-deg. pressure angle)

Q = Ratio factor $= \dfrac{2N}{N+n}$

N = Number of teeth in gear

n = Number of teeth in pinion

The above formula gives limiting load values and should be used with a safety factor ranging from 2 to 3. For all proportions and dimensions, the design of the gear will be determined by the formula which gives the lower allowable load for strength or wear resistance.

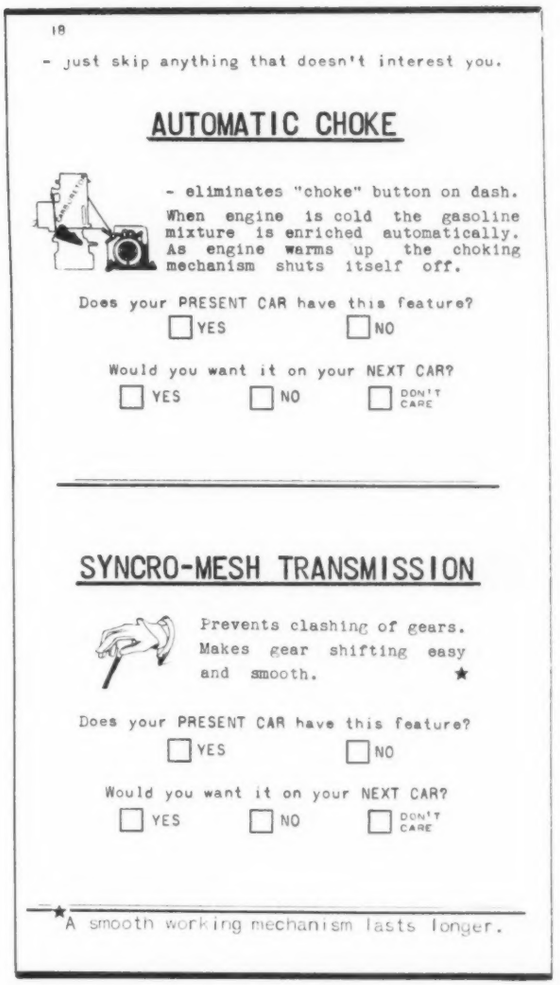
Metals to Withstand Sub-Zero Temperatures

Tests were made in the laboratories of the U. S. Army Air Corps to determine the tensile strength, hardness and other properties of ferrous and non-ferrous alloys at temperatures encountered during flight. The standard temperature of tests selected was —40 deg. C., which corresponds, approximately, to an altitude of 30,000 ft.; the materials investigated included five nickel-chromium stainless steels, a number of aluminum-base alloys and magnesium-base alloys.

The results indicated that the engineer-designer is safe in using, for aeronautical construction, permissible stresses as determined by testing the materials at normal temperature.

What the Customer Wants

Most engineers are awake to the importance of giving the customer what he wants. Those connected with General Motors companies have a really satisfactory way of finding out


18
- just skip anything that doesn't interest you.
AUTOMATIC CHOKE
- eliminates "choke" button on dash.
When engine is cold the gasoline mixture is enriched automatically. As engine warms up the choking mechanism shuts itself off.
Does your PRESENT CAR have this feature?
YES
NO
Would you want it on your NEXT CAR?
YES
NO
DON'T CARE
SYNCRO-MESH TRANSMISSION
Prevents clashing of gears. Makes gear shifting easy and smooth. ★
Does your PRESENT CAR have this feature?
YES
NO
Would you want it on your NEXT CAR?
YES
NO
DON'T CARE
★ A smooth working mechanism lasts longer.

A sample page from "The Proving Ground of Public Opinion"

YOU probably received one of those attractive little question booklets, "The Proving Ground of Public Opinion," sent out broadcast by the Customer Research Staff of General Motors. And the chances are that you went through it even though you may not have put down your answers and returned it. But did it occur to you as a designing engineer that this booklet is one of the most successful determiners of the customer's wants that has yet been devised? Or that the information brought back by it to the research staff and passed on to the designing department was invaluable to the engineers?

The time has gone by when the wasteful methods of high pressure salesmanship can be used profitably to "put over" a product on the buying public. When times were good and all of us were flush, what the salesman wasted was not so important, apparently. But now the market is so limited and profits are so hard to squeeze out that it is a vital matter to find out first what your customer will buy, and then design and build that instead of something you think he ought to want.

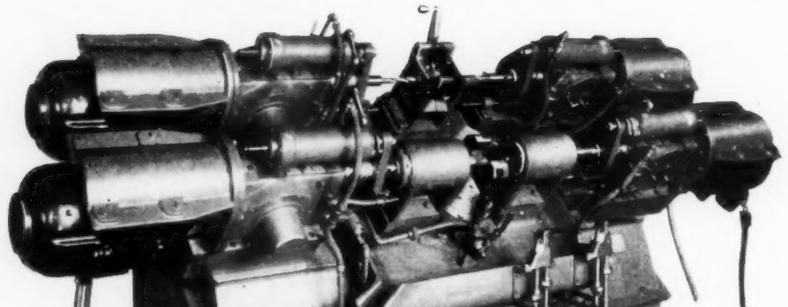
A very few engineers are so situated that they can talk with their customers and thus discover their desires. In such instances the product is likely to be highly specialized and the customers scarce. But where the general public is the customer it is out of the question to get the facts in this fashion. Here is where some kind of a questionnaire is essential.

Any old kind of a questionnaire will not do. Filling out such a document is more or less of a nuisance and your average citizen will not bother with it unless his interest is aroused. The man who undertakes its preparation must know something of human nature, psychology and the force of habit. The General Motors group made a careful study of all these factors before emerging with their question book.

But what will the engineer want to know about the customer's wants? If his product is a motor car he must, of course, be sure of his materials, of his design, of performance, of appearance. He must also know enough anatomy to reckon with nerve strain and with muscle tire. No matter how perfect a part is from the mechanical standpoint, if it subjects the driver or the passenger to discomfort, it is not what the customer wants, and is therefore not good commercial engineering practice.

When it comes to appearance the most expert advice on form and color may be at sea if it does not take into account the public fancy. In good times when long life is not vital, the product with the flashier appearance may meet with sales success. In times not so good the buyer thinks more of how long he is going to use what he buys and his preference runs to the more conservative design, the one that is not "dated" by some freak fashion.

All of this, and much more—suited to your own product, of course—can be dug up if your customer questionnaire is properly prepared. It must be interesting and it is usually wise to send it out in duplicate. Sketches and printed suggestions help to focus the attention of the prospect on the information you want to get from him. Make it as easy as possible for him to give you what you want, and you will be surprised how many people take an interest in your problem and undertake to help you solve it.



Dependable modern motors and controls are changing machines from single motor drive to unit motor drives, as in this Millholland piston boring, centering and drilling machine

Engineering the Unit Motorized Machine

WHERE power must be delivered to a number of units on a single machine, there is usually the choice of a mechanical drive from a common motor to the various units or an individual motor for each of the units. This second arrangement is generally modified to the extent that some of the motors may drive two or more closely associated units. It is this type of drive that is coming into great favor among designers of machine tools in particular and is receiving much attention in all fields of power-driven machinery.

Naturally, the first consideration when contemplating the feasibility of separate drives for each unit is the cost factor. Upon first thought it might appear that the separate motors and controls for driving each of the units of a machine would increase the cost considerably. But against the cost of the motors, wiring and controls must be credited the cost of the drive shafts, gears, clutches, bearings and other power transmission elements that are eliminated by the unit motor drive.

A factor that is not always given sufficient consideration when estimating the relative costs of these two types of drives is engineering expense. If only a few machines are to be built, the engineering cost per machine is likely to be greater than the cost of its manufacture. In the usual engineering procedure the problem is to devise means for furnishing power to the shafts or spindles that have been located in the machine. If a mechanical drive from a single motor is to be used it is then necessary to provide the required shafts, bearings, gears, chain drives, clutches and other elements that comprise the mechanical drive. Such a drive must be laid out with extreme care to be sure that there will be no interference. It is also necessary to allow for adjustments, provide for lubrication and furnish suitable safety guards or housings. This involves a great amount of detail work, all of which must be checked carefully. The cost of the necessary engineering calculations, layouts and drafting work mounts rapidly.

On the other hand, if the various units of the machine are to be driven by individual motors the whole design problem becomes greatly simplified and the engineering expense is correspondingly reduced. It is merely necessary to apply a direct-connected motor. If the spindle speed differs from the motor speed a "motor-reducer" unit can be used, or a simple gear reduction, belt drive or chain drive can be used. With such a compact arrangement there is little opportunity for interference and hence no great amount of time is required to check this item. The savings in engineering costs of the unit motor drive over the single motor drive will in many instances be greater than the cost of the drive for one machine.

Considering the relative maintenance cost and dependability of operation of these two types of drives, the unit drive has many advantages. The only attention required of a motor is the lubrication of its two bearings. And if these are of the anti-friction type, it is merely necessary to oil or grease them about once a year. In contrast to this is the almost daily attention

that must be given to power transmission elements. Bearings must be lubricated, clutches adjusted, chains tightened and other items must be given attention.

As to dependability, in a motor drive direct to the spindle, there are only two elements to fail. One is the motor and the other is the motor control device. The mechanical simplicity of both of these is such that if they have been selected and applied properly there is no reason for their failure during the life of the machine to which they are applied. Practically every breakdown in the electrical drive of a machine can be attributed to the motor being too small or too lightly constructed, control equipment whose mechanical design is too light for the service required, or failure caused by the absence of proper safety guards, such as limit switches, mechanical or electrical interlocks or protective relays.

Composed of a frame, stator, rotor and two bearings, it is hard to conceive of a power unit that is simpler mechanically than the induction motor. Barring accidents and abuse, such a motor will last indefinitely. But the desire to use the cheapest motor without proper regard to operating requirements will invariably result in trouble.

For a household device that is operated only a few hours a week it is good engineering to use a relatively light and cheap motor with small shaft and bearings. The average domestic washing machine is used a maximum of 3 hr. per week. Assuming that the machine is to have a life of ten years, the total actual running time would be less than 1,600 hr. The average production machine operates as many hours in less than six months and is expected to run continuously for many hours at a stretch.

It is folly in designing a machine that costs thousands of dollars to attempt to save a few dollars by using a motor that is cheaper but not sufficiently strong and powerful. And this applies also to all control and protective devices. It is not good enough to have everything function 99 per cent of the time. The automatic control that fails to stop the motor only once might wreck the machine.

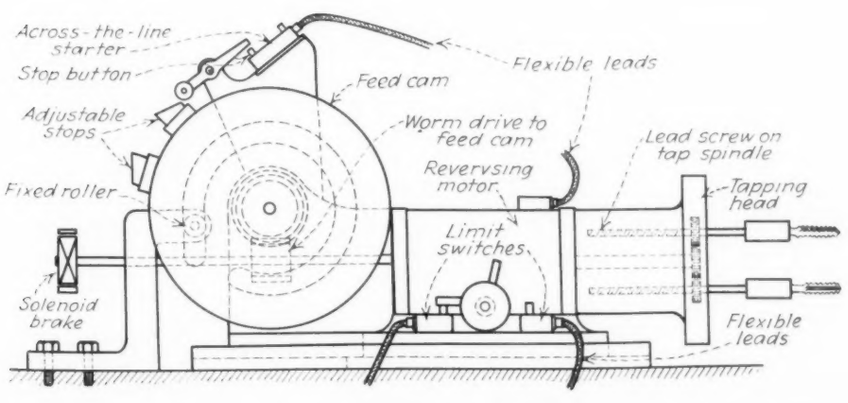
It must be admitted that every electrical automatic starter cannot be expected to operate indefinitely without once failing to function. Therefore, the careful engineer will provide some auxiliary safety device if such a failure would be likely to cause damage. As an example, the accompanying Fig. 1 shows diagrammatically a simple safety feature. Its application to a tapping machine is used for the purposes of the illustration. A reversing motor drives the tapping head, the stroke of which is determined by the cam actuated limit switches. The tapping head is moved up to and away from the work by a plate cam with a closed roller track. The roller is fixed on the bed of the machine and the cam rotates and travels with the head. The tapping head merely brings the taps up to the work, the taps then being fed by lead screws on each of the tap spindles. It is evident that should the motor fail to reverse and the taps continue to rotate in the same direction, much damage would be done. Such a condition might be caused by a failure in the pilot circuit controlled by the limit switches or by some mechanical failure in the starter or automatic control.

To eliminate the possibility of damage by the motor failing to reverse, two adjustable knock-off lugs are placed on the feed cam, as indicated. Should the motor fail to reverse, the continued rotation of the feed cam will cause one or the other of the knock-off lugs to press the stop button of the across-the-line starter, thereby opening the motor circuit and stopping the machine. Nor could the machine be restarted without first shifting the knock-off lug to permit pressing the start button of the across-the-line starter. This feature will naturally induce the operator to investigate before restarting the machine.

Another feature that is illustrated in this machine is the use of solenoid brakes. In the opinion of the designer of the machine, too little attention is generally given to the inherent possibilities of this simple little device.

Solenoids can also be used to throw clutches. For such a purpose, the solenoid has the advantage that the clutch will be thrown the moment that the electrical contact is made in the solenoid circuit. And to make the electrical contact requires only an extremely small movement. With adjustable contact points, they can be set to throw the clutch each time at exactly the same point in the cycle of operation of the machine.

There are quite a number of machines that use solenoid operated clutches and there are undoubtedly a great many more machines wherein they could be used advantageously. The "Casttype" machine, described in *Product Engineering*, January, 1933, uses a solenoid. The Heald grinder (*P.E.*, Dec., 1932, p. 492) uses such a control for reversing the table traverse. About two years ago solenoids came into use on automobiles for throwing the starting motor pinion in mesh with the flywheel gear. Numerous other examples could be cited.



Motor driven tapping unit. The cam-feed positions the unit, the taps are fed individually by lead screws on the tap spindles, the limit switches control the reversing motor, stops on the feed cam opening the starter switch in case the limit switches fail. The solenoid brake prevents coasting

Anti-Friction Bearing Mountings for Heavy-Duty Spindles—I

F. A. FIRNHABER
Designing Engineer, Landis Machine Company

Typical proven designs of bearing mountings for spindles that are subjected to heavy loads and run at medium speeds

FIG. 1

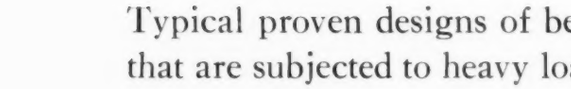
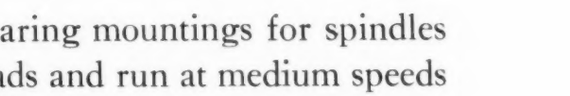


FIG. 2



IN THE design of low and medium-speed spindles mounted on anti-friction bearings the problems is somewhat simplified as compared to high-speed spindles. There is no danger of the spindle whipping because of dynamic unbalance. (See *P. E.* Sept. and Oct. 1933, "Ball Bearing Mountings for High-Speed Spindles"). But other than that the problem is quite similar. The possibility of axial play must be avoided by using preloaded bearings or automatic axial adjustment by spring pressure or by manual adjustment. Both the spindle and bearing housings must be rigid, all machining must be done accurately, and provision must be made for lubrication, the retention of the lubricant in the housings and the exclusion of dirt.

Selection of the size of bearing required is usually a relatively simple matter. The only danger here is the possibility of a mistake in calculating the magnitude of the loads that might be imposed on the bearings. But if a bearing of adequate size is selected on the basis of the manufacturer's catalogs and recommendations there will not be much chance of failure because of small overloads on the bearing. Adequate bearings will assure minimum bearing friction, accurate running of the spindle and a long bearing life.

The typical heavy-duty slow-speed spindles shown here are designs that have been developed with the aid of the engineering department of the bearing manufacturer. The name of the latter is given in each instance.

Fig. 1—This is a design of spindle mounting for a horizontal milling machine using a special form of end mill, approved by the Gurney Ball Bearing Company. The end and radial thrusts being comparatively light, the spindle is carried on two deep-groove ball bearings. The spindle *A* holds the cutter chuck *B* by means of the draw-back rod *C*. There is no "float" in the spindle as the principal load is end thrust, which is taken by the rear bearing whose outer race rests against the thrust flange *D*. This flange is accurately finished and fitted so that its shoulder height will not permit axial play or cause binding of the spindle when the flange has been screwed into place. One nut *E* clamps both bearings, oil retainers and the long pinion *F* driven from gear *G*, the whole assembly feeding forward past the gear. Maximum speed of this spindle is 2,000 r.p.m. The spindle is shown in its forward position.

Fig. 2—This spindle assembly is of the same type as that shown in Fig. 1, and is also mounted on Gurney ball bearings. It is designed for larger cutters, for which reason two deep-groove bearings are provided at the front to take the increased radial load, while the rear bearing is of a type having greater capacity for end thrust than for radial thrust. Felt rings *A*, pressing lightly against the spindle, retain the lubricant. The knob assembly *B* locks the spindle against rotation when tightening or loosening the cutter chuck with the draw-back rod.

Fig. 3—In this unusual design of the spindle assembly for a small automatic milling machine, the S.K.F. ball bearings are always under an axial load exerted by springs. There is no possibility of axial play in the spindle. The steel spindle *A*, made in one piece with the pinion *B*, is mounted on two deep-groove bearings. Two springs sliding on the spring rod *C* are diametric-

ally opposite each other in a horizontal plane. The rod head and the supporting bushing *D* are pressed by the springs against the outer bearings, compelling contact between the balls and races at all times. The front bearing, to the right, is retained by the pressed-on collar *E*. A felt seal in this collar protects the bearing from dirt. The rear bearing is clamped in place by a threaded collar *L*, with the hardened and ground bushing and collars *G* and *H* taking the end thrust. Drive to the spindle pinion is from the gear *J*. This spindle is designed for a maximum speed of 1,000 r.p.m. Collars *K* retain the lubricant in the bearings.

Fig. 4—Four S.K.F. bearings are used in this mounting for a vertical milling machine spindle. The spindle proper, *A*, rotates in two deep-groove ball bearings inside the sliding sleeve. Upper bearing carries all of the cutter end thrust and transmits it to the flange *D*. The outer race of bearing *C* has a sucking fit in the sleeve, and in the lower sleeve the cap *F* houses a series of helical springs that exert pressure against the outer race of the lower bearing which supports the weight of the spindle proper. This automatically takes up all axial play. Clamping nut *E* presses against the inner race of the upper bearing and transmits the pressure through the spacing sleeve, thereby pressing the inner race of the lower bearing against the flange at the bottom of the spindle.

The driving pulley *K* is keyed to the quill *L*, which is supported on two self-aligning ball bearings. Belt pull has no effect upon spindle deflection. These bearings are mounted in the stationary sleeve *M* bolted to the frame of the machine. The spindle is driven by the key *N*, which is free to slide in the spline in the quill *L*. Oil introduced at *P* passes through a screened felt strainer *R* and then feeds by gravity from one bearing to the next, finally draining out of the sleeve cap *F* at the bottom of the spindle.

Fig. 5—Four S.K.F. preloaded deep-groove ball bearings support the milling machine spindle shown here. All the end thrust and heavy radial loads are taken by the large bearings at the right end whose inner and outer races are held in a fixed position on the spindle and in the housing respectively. The rear bearings are mounted in a stationary flange bushing *A*,

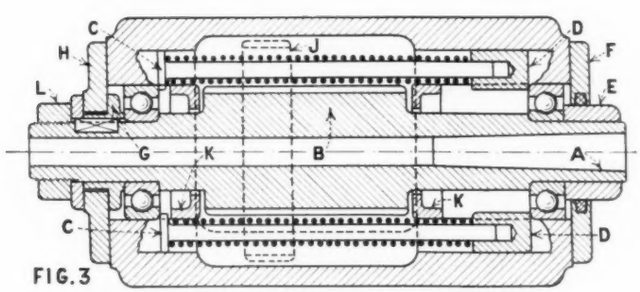


FIG. 3

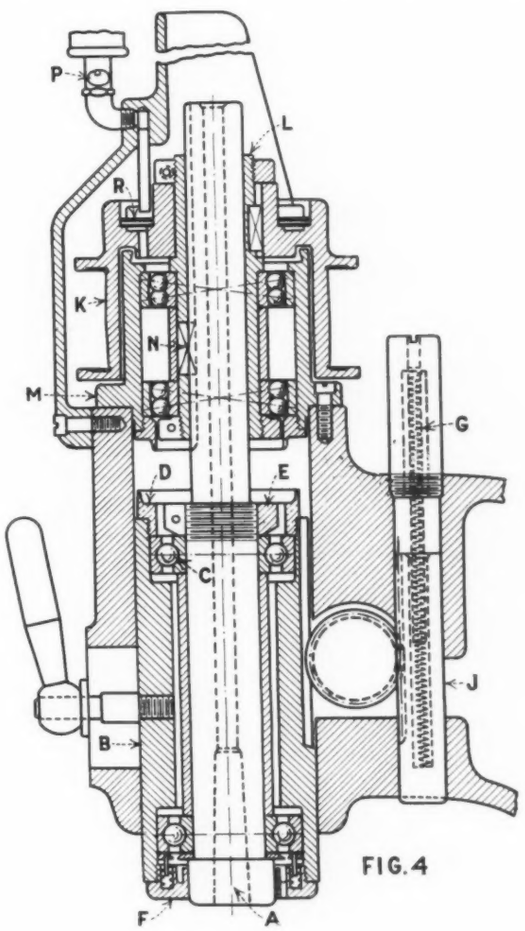


FIG. 4

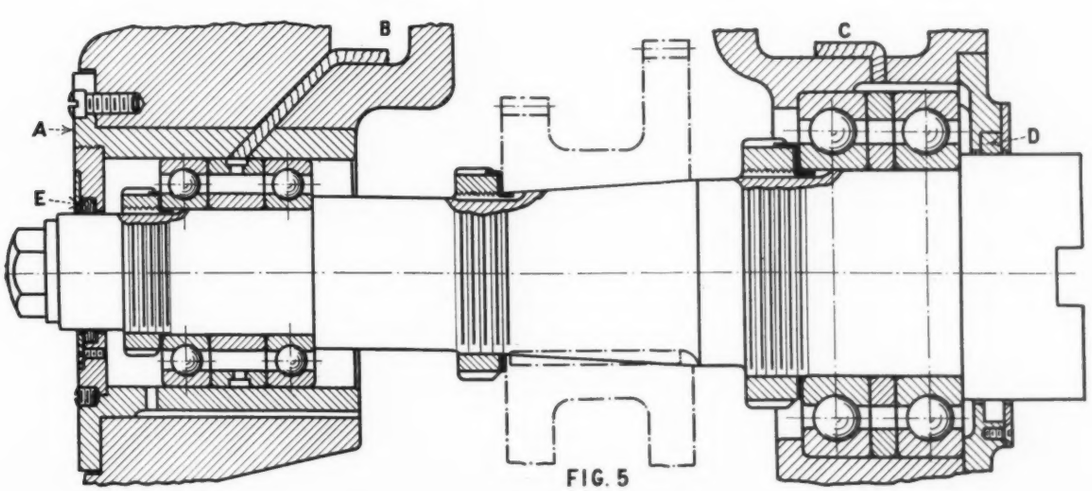


FIG. 5

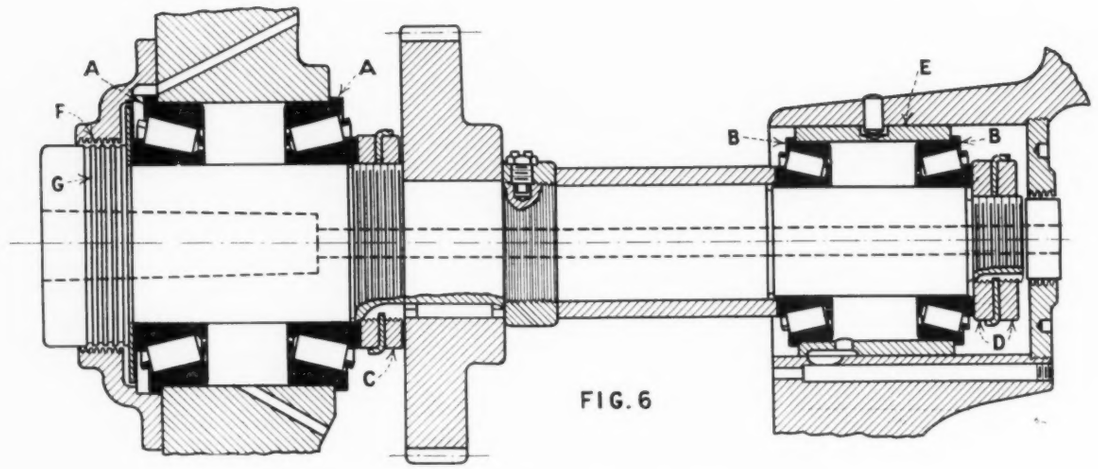


FIG. 6

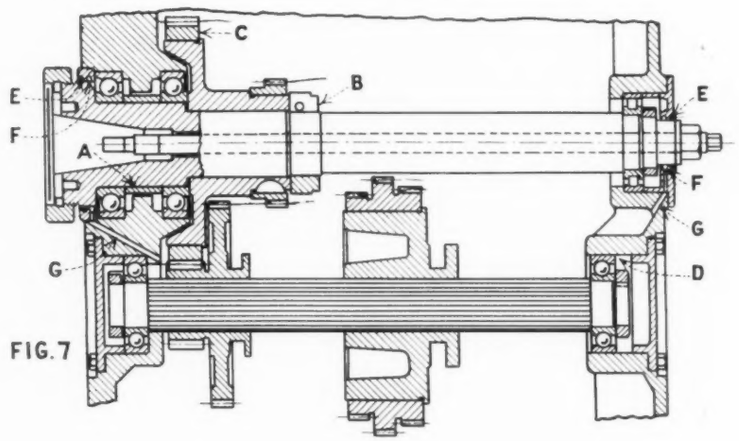


FIG. 7

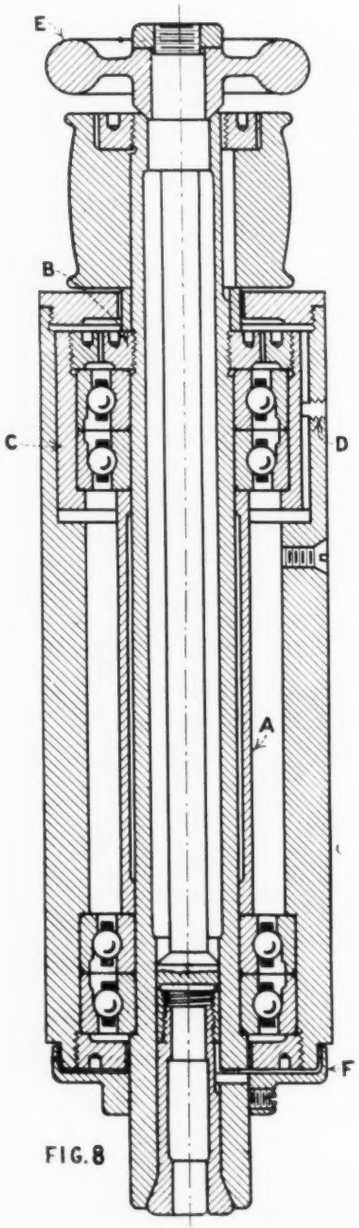


FIG. 8

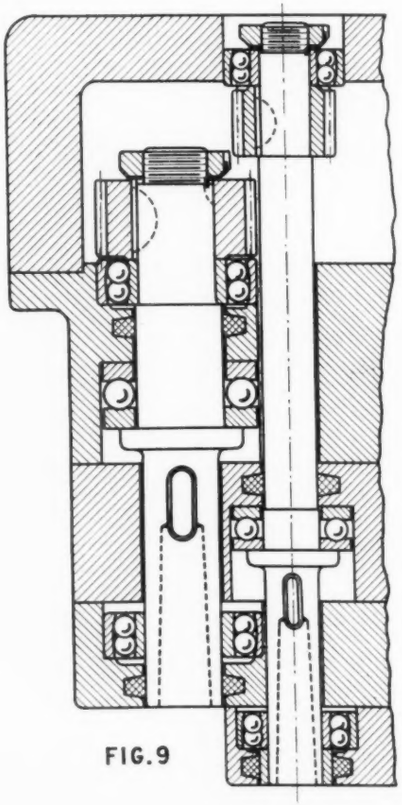


FIG. 9

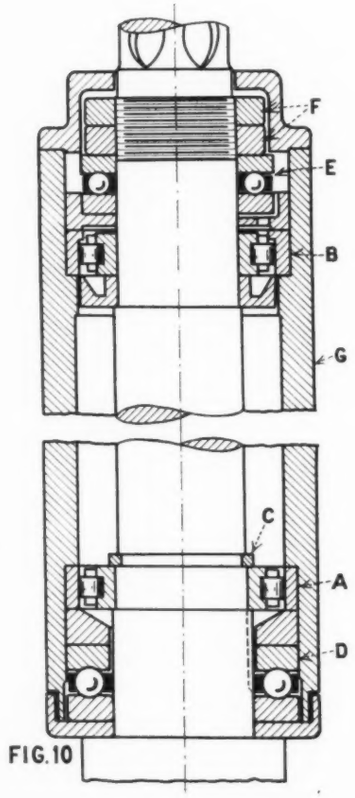


FIG. 10

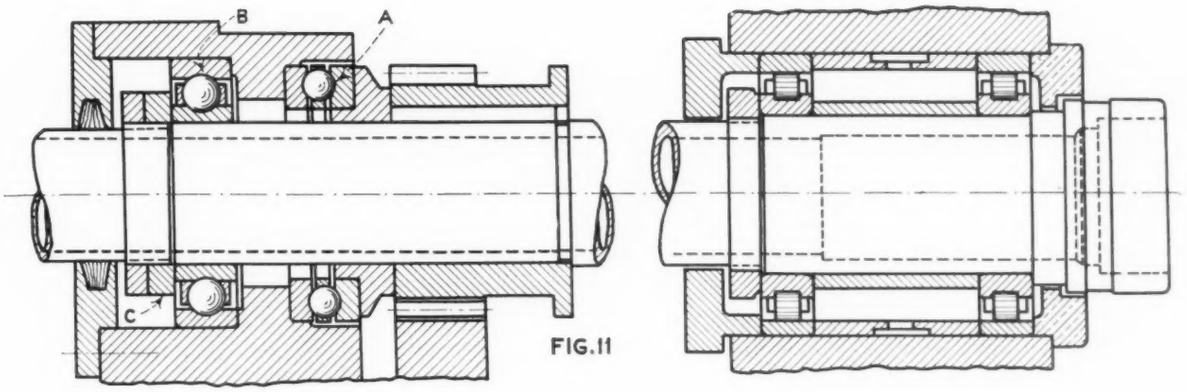


FIG. 11

the outer races of the bearings being a floating fit in the bore of the bushing. This allows for any longitudinal expansion caused by temperature changes. From the pockets *B* and *C* oil is supplied to each bearing through felt wicks. The seal at *D* is a diagonally split brass ring retained by a steel plate, while a cork seal *E* is used at the rear end.

Fig. 6—Spacers or holes bored to two different diameters are not needed when using bearings with flanges on the outer cups. In this design, the Timken bearings are adjusted by the spindle nuts *C* and *D* which keep the flanges *A-A* and *B-B* in contact with the finished end faces of the bored holes. To take care of expansion and contraction the rear bearings are mounted on a slidable sleeve *E*. The series of grooves at *F* and *G* keep out chips and dirt. A splash system supplies the bearings with lubricant.

Fig. 7—Both roller and ball bearings are used in the main spindle shown here equipped with Norma-Hoffmann bearings. The length of the spacer *A* between the inner races of the two ball bearings is made slightly short so as to set up a specified preload when the bearing and gear assembly is clamped with the nut *B*. Because of space limitations the bearings are put close together. Conventional practice would require about 50 per cent more space between them. However, in this design close spacing is permissible as the slow-speed gear *C* which carries the greatest tooth load overhangs the bearing only slightly.

Endwise expansion of the shaft is taken up in the roller bearing at the rear. This roller bearing takes only radial loads, the rollers being free to slide axially in the outer race. Therefore it is not necessary to allow clearance between the outer bearing race and the housing. On the other hand, for the rear bearing of the countershaft at *D*, the entire bearing must be free to move axially to allow for expansion. Felt seals *E* at each end of the main shaft protect the bearings against dirt. Lubricant is supplied to the bearings by a splash system and is returned to the gear box by the inner grooves through drilled holes.

Fig. 8—Open-type Norma-Hoffmann ball bearings selected and matched face to face are used in pairs on this vertical router spindle. The inner races of both sets of bearings are pressed on the spindle, spaced by the separating sleeve *A* and clamped by the nut *B*. Also the outer races of the bearings are clamped against the shoulders in the sleeve. All end thrust is taken by the lower pair of bearings, the upper pair being mounted in a slidable sleeve *C* free to shift axially with the expansion or contraction of the spindle. To prevent rotation of the movable sleeve, a pin *D* in the outer sleeve engages in a keyway in the sliding sleeve. The handwheel *E* controls the drawback rod which holds the tool in the chuck at the lower end of the spindle. The slinger *F* is used to guard against the entrance of dirt.

Fig. 9—In multiple-spindle drill heads where short center distances are involved, the bearings can be staggered. In this design, S.K.F. self-aligning bearings are placed as close as possible to the imposed loads and take the gear thrust. This assures the smallest possible drill eccentricity. Heavy ball thrust bearings take the upper thrust of the drill.

Fig. 10—In this radial drill spindle mounted on Norma-Hoffmann bearings, the radial load is carried by cylindrical roller bearings *A* and *B*, both bearing against shoulders in the sleeve. The end thrust in both directions is taken by heavy-duty ball thrust bearings *D* and *E* which carry no radial load. Upward thrust is all taken by the lower thrust bearing, necessarily of a larger size than the upper one which receives only occasional downward thrust when back boring or facing. The entire lower bearing assembly is retained by a square wire snap ring *C* which makes it unnecessary to thread the spindle for a nut. Endwise adjustment for both upper and lower thrust bearing is controlled by the nut *F* at the top. To guard against dirt, the top bearings are capped while the lower ones have an overlapping tongue seal. The upper end of the spindle is splined for the driving pulley and the whole assembly can slide in its housing.

Fig. 11—On automatic machines where four or five closely spaced spindles are used, cylindrical roller bearings of extra light type with large bores and small outside diameters can be pressed and clamped on the spindle at the tool end. In this design using Norma-Hoffmann bearings, the ball thrust bearing at *A* takes all endwise thrust of the working load and a deep groove single-row ball bearing at *B* carries the thrust in the opposite direction and locates the spindle. This bearing also carries the radial load at the rear. The inner ring of the bearing is adjustable endwise by means of nut *C* which takes all play out of the spindle.

COMMUNICATION and COMMENT

Variable Speed Transmission

H. ANTHONY NELSON
De Kalb, Ill.

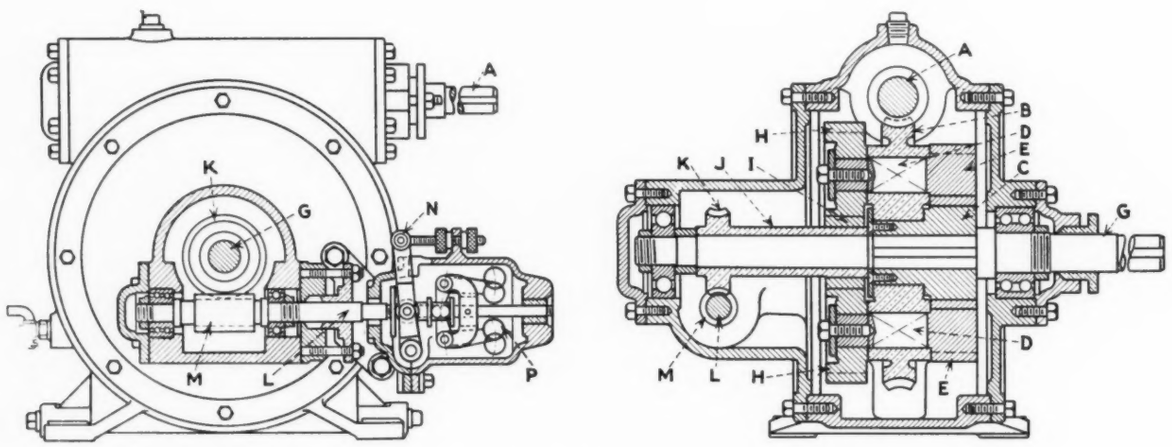
Combining a differential gear unit with an adjustable escapement the variable speed transmission, shown in the accompanying drawings, by-passes a variable portion of the drive shaft revolutions, thus varying the speed of rotation of the output shaft. Power applied to the worm shaft *A* drives the worm *B* which is free to rotate on the hub of gear *C*. The worm gear carries two pinion shafts *D*, to which are attached the fixed pinions *E*. Thus, motion of the gear *C*, which is attached to the output shaft *G*, is obtained entirely from the pinion *E*.

Fixed to the other end of the shaft *D* are gears *H*, free to rotate around the pinion *I*. Thus, the pinion shaft *D* receives its motion from rotation of the pinion *I*, which is a part of sleeve *J* carrying the worm gear *K*. This assembly is free to rotate on the main shaft *G*, but rotates only as fast as the worm *M* allows.

In the gear train thus far described, power applied to the worm shaft *A* will rotate the worm gear carrying with it the pinion shaft. Rotation of the pinion *H* around the pinion *I* causes the shaft *D* to impart motion to the output shaft *G*. The rotation of the worm shaft *L* is controlled by an escapement. The worm shaft driven by the worm wheel carries governor weights *P* with an adjustment at *N* which varies the position of fly-ball weights. By adjusting the position of the weights the speed of the governor shaft is varied in turn controlling the speed of the output shaft.

From tests on a model of this transmission a speed reduction ratio from 7.4 to 10 was obtained under a constant prony brake load of $1\frac{1}{4}$ lb., the reduction varying with the adjustment of the governor. With a brake load of $1\frac{5}{8}$ lb. the gear reduction ratio was from 4.7 to 5.2. Efficiency of power transmission under this load varied from 61 to 53 per cent as the ratio was increased.

Very little power is lost in the escapement and it is noted that the power from the prime mover by-passes the escapement to the output shaft, which permits ease of adjustment under full load. The governor also maintains a steady operating r.p.m. at the output shaft. The operating efficiency is satisfactorily high and decreases in proportion according to the reductions in r.p.m. at the output shaft. Different gear ratios incorporated in the differential and the elastic design of the escapement, present an unlimited selection of reductions and variations.



When under constant load, the speed of the output shaft *G* is varied by adjusting the position of the centrifugal weights *P* in this variable speed transmission

Furniture Ornaments

To the Editor:

Ornaments of the nature described by C. F. Scribner, in the November number of *Product Engineering*, were often used in times past, but usually were bought from firms specializing in their manufacture. The use of this type of ornament is decreasing, although there are still several firms who produce them on order. For low-priced decoration, engineers usually turn to decalcomania transfers, except, of course, when decoration must be in relief.

For low-cost relief decoration on large quantities embossing is often used, both on sheet metal and also on

wood parts for furniture. Two types of machines are available for embossing on wood; one is a toggle-press machine which produces clear-cut impressions with heated dies on such woods as black gum. In the other type of machine the surface to be embossed is passed under a die cut in the face of a roller.

For ornaments used on better-class wood furniture, modern carving machines have multiple spindles connected to a master spindle which is guided by a tracer tool over a master pattern. For high production work right hand and left hand patterns are made from a single pattern, half of the spindles producing rights and the other half lefts. —JOHN E. HYLER
Peoria, Illinois.

How Should Limits Be Shown?

To the Editor:

A subject that has caused considerable discussion is the marking of limits on drawings. Dimensional tolerances can be written in the four following ways:

$$0.250 \pm 0.002$$

$$0.252 \begin{matrix} + \ 0.000 \\ - \ 0.004 \end{matrix}$$

$$0.248 \begin{matrix} + \ 0.004 \\ - \ 0.000 \end{matrix}$$

$$0.248 - 0.252$$

All these ways of expressing the limits about the basic dimensions 0.250 would give the same results in factory practice. The last manner expressed, namely 0.248 — 0.250, I have found to be the most practical as it saves the possible chance of the man in the factory making a slip in his figuring. I have found others recently, who insist that one of the other three methods are preferable. Personally, I believe that all dimensions should be given in such a way that the workman has no figuring to do. I would like to hear other opinions upon this subject.

—R. E. HOLBROOK
Attleboro, Mass.

Sonic Non-Destructive Testing

E. J. TANGERMAN
Assistant Editor, Power

A recent German publication describes a method of testing structures that will undoubtedly prove of distinct value in the solution of many engineering problems. The principles are not entirely new as this particular method of testing has been used previously in isolated instances. The method is simple and requires but little equipment and is particularly applicable to beams under tension. For large structures, the tests can be made on scale models. In principle, it operates by the comparison of sound waves.

To determine the load on a member, two clamps carrying uprights are fastened at a given distance apart on the section to be tested under tension. A piece of piano wire is stretched between the two uprights so that it gives forth an audible note when it is vibrated. An exactly similar section of piano wire is then set up nearby, being stretched from an upright of the same type and same distance apart, the tension in this second wire being obtained by means of a suspended weight. By increasing or decreasing the tension weight, the second wire is carefully tuned to the same musical note as the test wire.

As the tension in the beam being tested increases its elongation will cause a change in the pitch of the sound coming from the wire stretched between the uprights. For a given gage and length of wire the natural frequency will vary with the stress. With a given load on the test beam the comparison wire is brought up to the new pitch by loading on additional weight. Since the moduli of elasticity of both wires are the same and also the same as that of the beam (since all are steel) the stress in the wire on the beam can be figured readily by the formula $L/A = S$, where L is added loading in pounds, A is cross-sectional area in square inches, and S is stress in lb. per sq.in.

This method of testing has found numerous applications where it was desired to determine the increase in stress resulting from an increase in the load on the structure. In order to eliminate possible errors that may arise as the result of a mechanical twanging, the strings are vibrated magnetically.

By applying a correction factor determined by the ratio of the moduli of elasticity of the two different materials, steel and whatever the material of the beam may be, the stresses can be calculated readily when the beam is other than steel. This method of testing was used extensively in the airships Akron and Macon to determine the load on the structural frame resulting from wind and shocks. Similarly, it is finding considerable application of airplane frames or fuselages for the determination of stresses in flight.

At Princeton University, W. R. McNamee, under the direction of Prof. C. H. Willis, developed a special musical tensometer. It has an electromagnetic system for inducing a variable frequency of vibration in the wires and a telephone circuit for picking up the tone given off by the wire.

Photo-elasticity for the Designer

G. B. KARELITZ
Associate Professor of Mechanical Engineering
Columbia University

In designing machine details, photo-elasticity is a tool of real service, yet the work of most photo-elastic laboratories is confined to checking mathematical solutions of stress distribution problems. This laboratory attitude has obscured, to some extent, the effectiveness of photo-elastic analysis in disclosing sources of fatique failure, particularly during the early stages in the engineering development of a machine.

Where weight and space are at a premium and stresses are necessarily higher than permitted by the usual safety factors, actual stress distribution is much more important than the stresses indicated by theoretical analysis. Knowledge of actual stress conditions is of particular importance under two conditions: when the load is variable, and when the cross-section is variable. For a study of stress distribution under these two conditions polariscopes suitable for use in design work can be obtained for less than $200. With equipment available at this low cost the photo-elastic method of stress analysis is, therefore, available to even the smallest engineering department.

In addition to the series of articles published in *Product Engineering*, the photo-elastic laboratory of Columbia University has prepared reports on the following investigations which may be of interest to product engineers. Reprints of these papers are available.

"Photo-elastic Study of Shearing Stresses in Keys and Keyways," A. G. Solakian and G. B. Karelitz, *Trans. A.S.M.E., Applied Mechanics,* June 15, 1932, pages 97-123.

"Photo-elastic Study of Shearing Stresses in Dardelet Thread and Nut Connections," A. G. Solakian and A. H. Beyer, Report No. 2287, Testing

Laboratories, Department of Civil Engineering, Columbia University.

"Graphical Representation of Two-Dimensional Stress Distributions," A. G. Solakian, *Journal of the Franklin Institute,* August, 1933, pages 217-224.

"Photo-elastic Analysis of Stresses in Composite Materials," A. H. Beyer and A. G. Solakian, *Proc. A.S.C.E.,* September, 1933, pages 1121-1132.

"Stresses in Transverse Fillet Welds by Photo-Elastic Methods," A. G. Solakian, *Journal of the American Welding Society,* January, 1934.

Determining the Base of the Triangle

In closing the discussion of the problem submitted by C. W. Hinman in the October number, both graphical and analytical solutions are given here. All of these methods give two base dimensions of the triangle satisfying the problem—that is, to find the base of an isosceles triangle with 20-in. sides, circumscribing a 9-in. square. The three analytical solutions agree to the third decimal place, the shorter base dimension agreeing with several solutions given in the December number.

To the Editor:

Trigonometric relations based on the data given in Mr. Hinman's problem give the following two equations:

$$(x + 4.5)^2 + (y + 9)^2 = 400;$$
$$x/9 = 4.5/y$$

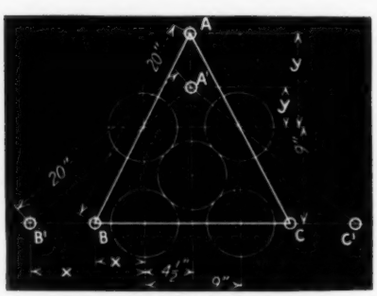
Solving these equations simultaneously gives the following fourth power equation in x:

$$x^4 + 9x^3 - 298.75x^2 + 729x + 1640.25 = 0$$

Since this is a fourth power equation there will be four roots, but in this problem only the positive roots are needed. Applying Descartes' Rule of Signs it is apparent that there are not more than two positive roots, and that there are two negative roots. Hence, there are two and only two positive roots.

From a scale drawing of the problem it is apparent that the two positive values of x will be between 4 and 5 and between 10 and 11. Using Horner's method of synthetic division, the two roots to the nearest thousandth are found to be 4.5961 in. and 10.9518 in. Thus the base of the triangle is either 18.1922 in. or 30.9036 in.

—Sylvester S. Snyder
Landis Machine Company.

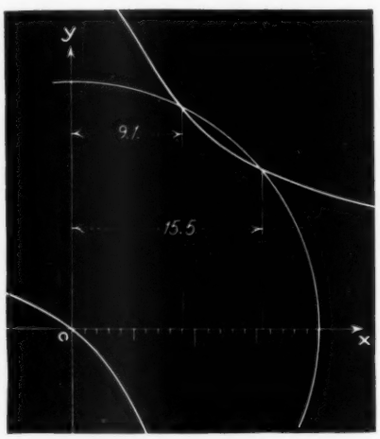


Synthetic division is used by Mr. Snyder to solve the fourth power equation giving the dimension x

To the Editor:

From the comparison of the whole triangle to the small triangle above the square the following two different equations in y can be obtained.

$$y = \sqrt{400 - x^2};\ y = 18x/(2x - 9)$$



When the two equations in y are plotted their intersections indicate values of x that satisfy the conditions of the problem

A solution of the two simultaneous equations for x and y involves an equation of the fourth degree, which is difficult to solve. A solution can be obtained graphically by drawing the curve of the two equations in y and determining the intersection of the two curves. Thus, the first curve will be a circle with a radius of 20, and the second curve will be a hyperbola, as shown in the accompanying figure. From the approximate point of intersection of these two curves, exact values for x are obtained by cut and try methods. This method gives 18.20 in. and 30.90 in. as the two base dimensions satisfying the conditions of the problem—Norman C. Harris
Los Angeles, Calif.

To the Editor:

From the relation given in the problem the following equation in terms of x is obtained:

$$x^4 - 9x^3 - 298.75x^2 + 3{,}600x + 8{,}100 = 0$$

From this equation it is apparent that there are two real roots and two complex roots. The real roots can be obtained directly to any degree of accuracy by using Graeffe's method, but this involves raising the various terms to 32d or higher powers and involves an accurate use of logarithms.

A solution can be obtained more readily by plotting the following two curves:

$$y = \frac{81x^2}{x^2 - 9x + 20.25};\ y^2 = 400 - x^2$$

Approximate answers are given by the intersections of these two curves, and the desired accuracy is then obtained by cut-and-try methods. Solutions obtained in this manner give base dimensions of 18.1928 in. and 30.9004 in.

—H. F. Bariffi
Hamden, Conn.

To the Editor:

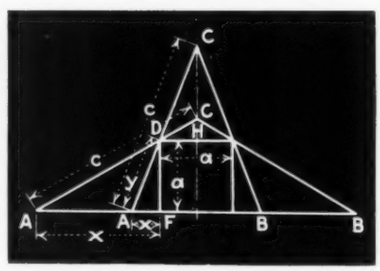
Referring to the figure the right triangles ADF and DCH are similar, so that $DC = AD \cdot DH/AF$. Therefore,

$$c = \sqrt{a^2 + x^2}\,(1 + a/2x).$$

Simplifying this relation gives the general equation:

$$4x^4 + 4ax^3 + (5a^2 - 4c^2)\,x^2 + 4a^3\,x + a^4 = 0$$

Substituting in this general relation the values for a and b given in the problem and applying Descartes' Rule of Signs it is found that this biquadratic equation has two positive and two negative roots. The negative roots of course must be disregarded in this problem. Applying Horner's method for incommensurable roots, the two values for x are found to be



A general equation for the distance x is developed by Mr. Johnson

4.59610 and 10.9519. The length of the base AB of the isosceles triangle is then 18.1922 in. and 30.9039 in.

—William Johnson
Cleveland, Ohio.

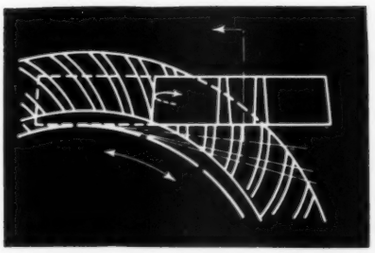


Fig. 2—Fellows cutters can also be arranged to cut spiral crown gears rotating both cutter and blank as shown by the arrows

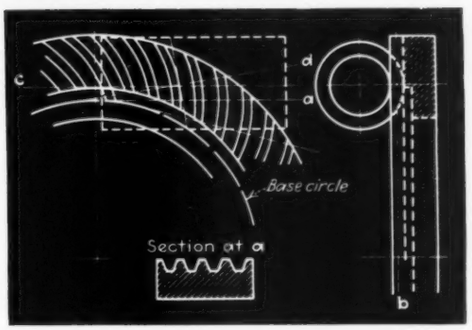


Fig. 1 — Spiral crown gears generated with a hob on Pfauter machines have involute rack section teeth in planes parallel to the gear axis

5. Drafting with attention to designs fitting in with established standards, yet permitting variations and special designs at low cost.

6. Preliminary samples to check the designs.

7. Final development based on tests and revisions of samples.

8. Cost analysis including consideration of tools and fixtures, machines, materials and methods.

9. Preparation of engineering information such as test specifications, instruction for use, photographs and sales data including application and special features of the design, tabulated information on performance, limitations, weights, etc., field follow-up of early production for improvements. —L. J. LUNAS
Newark, N. J.

To the Editor:

In the following equation, obtained from the conditions given with Mr. Hinman's problem, x is the distance along the base of the triangle from the gear center to the nearest guide pin:

$$x^4 + 9x^3 - 298.75x^2 + 729x + 1640.25 = 0$$

Using Horner's method for obtaining a solution to this equation, two positive roots for x are found, 4.596 in. and 10.952 in. From these values the two base dimensions that satisfy the conditions of the problem are 18.192 in. and 30.904 in.—C. D. MACY
Austin, Minn.

Development Procedure

To the Editor:

In regard to the article in the November *Product Engineering*, "Getting a New Product Under Way," none of the four development procedures give the essential steps in the engineering development. The following outline, confined to engineering development procedure, undoubtedly applies to almost any product, although the steps are based on the development of electrical instruments.

1. Research investigation of general and specific problems.

2. Preliminary development to determine the general lines of the design.

3. Working models confirming theory and improving performance.

4. Design of parts and selection of materials and finish consistent with cost, manufacturing equipment and personnel, and standardization.

The Involoid Gear

To the Editor:

Several alternatives to the lateral worm gear design described on page 393 of the October *Product Engineering* have been developed in England. For example, the Involoid gear, an invention of the writer, is a crown gear having teeth of involute rack section both in planes parallel with the gear axis and also in the plane normal to the axis. As shown in Fig. 1, when generated on a standard Pfauter machine by a standard 20-deg. hob with its axis at c set so the line d, normal to the pitch helix, is tangent to the base circle e, the teeth produced are so straight that error cannot be detected by ordinary means of measurement.

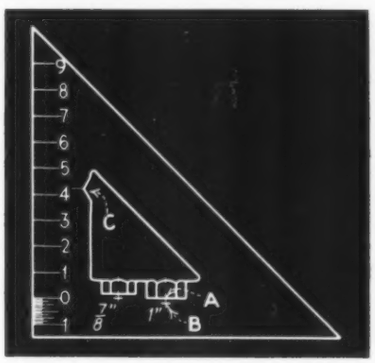
In the production of these special gears no special equipment is used. The hob is merely sunk vertically into the blank, and both rotated in unison, as a worm engaging with a wheel. Cast iron gears, 8-in. outside diameter, 1½-in. face, and 8-diametrial pitch are generated in about six minutes. Normally, the teeth produced are all of the same thickness at right angles to the tooth surfaces, and are of parallel depth, but a special angular attachment, which rolls the blank in conjunction with the hob and about an imaginary crown gear supposed to be rotating in unison with the hob, produces conical gears in the same machine.

These gears can also be cut with a standard Fellows cutter, as shown in Fig. 2. The cutter is mounted on the hob spindle, and the blank attached to the table in the usual manner. The cutter and blank are rotated together in directions shown by the arrows, the blank being fed along the base circle tangent and the cutter fed to the tooth depth. —FRANCIS W. SHAW
Heywood, England

Time Saving Triangle

J. S. CARPENTER
Detroit, Mich.

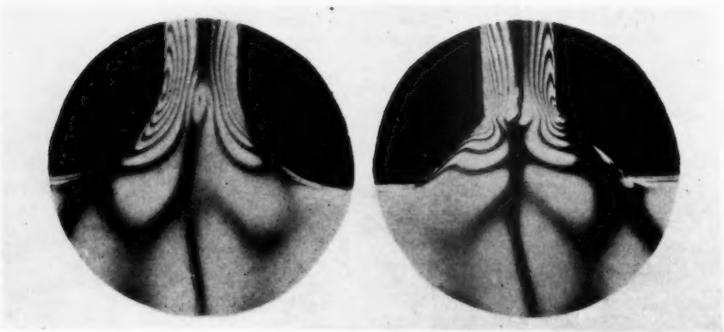
Time studies on drafting work show that approximately the same amount of time is spent in reaching for a scale and triangle as in the actual work of drafting. Nearly 25 per cent of drafting time can be saved in straight layout work by having the scale on the



A considerable saving in drafting time is obtained from the use of special scales and layouts on this 45-deg. triangle

triangle edge, as shown in the illustration. The vertical leg upon which the light strikes carries a scale divided into sixteenths, carefully scratched in so that thirty-secondths can be estimated easily. As an additional shortcut, curves are cut into the edge of the inside opening of the triangle for the commonly used American Standard bolt heads. The bottom of the nut is marked by a gage point A, and the bottom of the head by B. In laying in the bottom of tapped holes, the angle of the drill point, cut in at C, also saves time.

· NEWS ·



Closely spaced fringe lines at the sharp corners of the triangular fillet (right) indicate possible sources of fatigue failure, which condition is not present when curved fillets (left) are used

Applying Photo-Elasticity

Strengthening a welded joint by the removal of metal was demonstrated by Everett Chapman, vice-president in charge of engineering, Lukenweld, Inc., in a paper before the American Society of Mechanical Engineers on December 6.

The triangular throat weld, as shown to the right in the illustration, while having a lower average stress because of its greater throat dimension, has higher maximum stresses caused by the abrupt change in contour. The concave throat weld has a smaller throat area but the maximum stress is considerably lower because of a more uniform stress distribution.

In a discussion of Mr. Wiebel's paper on "Stress Concentration at Fillets," A. G. Solakian, whose work is already familiar to readers of *Product Engineering*, demonstrated that all stress concentration in a fillet is eliminated by the use of the spiral curve, although it was recognized that this curve would be difficult to reproduce exactly.

Simplified Twist Drill Sizes

A 59 per cent reduction in the number of standard twist drill sizes, representing a total reduction of 160 separate sizes, is proposed in a list submitted by the Sectional Committee on Small Tools and Machine Tool Elements, working under the procedure of the American Standards Association.

From $\frac{1}{64}$ to $\frac{1}{8}$ in., 24 drill sizes are proposed for each $\frac{1}{64}$. Diameters from $\frac{1}{8}$ to $\frac{3}{8}$ in. are covered by 12 sizes; from $\frac{3}{8}$ to $\frac{5}{8}$ by 15 sizes; from $\frac{5}{8}$ to $\frac{7}{8}$ by 16 sizes; and 3 sizes for each $\frac{1}{64}$ in. increment for diameters from $\frac{7}{8}$ to $\frac{63}{64}$ in. From $\frac{1}{32}$ to $\frac{1}{2}$ in. inclusive, the drill sizes advance by $\frac{1}{64}$ in.

American Standards Association Elects New Officers

At its annual meeting in New York on December 13, the American Standards Association re-elected as its president Howard Coonley, president of the Walworth Company; F. E. Moscovics was elected vice-president; J. C. Irwin, valuation engineer, Boston & Albany R.R., and chairman of the standardization committee of the American Railway Engineering Association, was elected chairman of the American Standards Council to succeed Cloyd M. Chapman. S. M. Farmer, chief engineer, Electrical Testing Laboratories, New York, was elected vice-chairman. The principal topic of discussion at the meeting was the consideration of the new problems incident to the broadened scope of the organization brought about by cooperation with the Bureau of Standards.

An Industrial Art Exposition

In the R.C.A. Building, 30 Rockefeller Plaza, New York City, and probably in the plaza outside, an exposition of industrial art will be staged from February 15 to March 14. It is being organized and directed by American designers, the various exposition committes, including such names as Teague, Dreyfuss, Bel-Geddes, Bernhard, Nash, Switzer, Van Doren, Arens, Schoen and Wright. The business of the exposition will be handled by the National Alliance of Art and Industry, whose offices are at the same address. Alon Bement is in charge for the Alliance.

The only eligibility restriction is that the object exhibited be one in which design is an element and which is intended for mechanical duplication, already produced or in process of production. Exhibit space costs $1.10 per sq.ft., and there will be an entrance charge of 25 cents. It is well within the bounds of possibility that out of this exposition will come a new American style of design.

MEETINGS

Society of Automotive Engineers—Annual motor show dinner, Jan. 8. Annual meeting, Book-Cadillac Hotel, Detroit, Mich., Jan. 22-25. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

American Institute of Electrical Engineers—Winter convention, Engineering Societies Building, New York, N. Y., Jan. 23-26. H. H. Henline, secretary, 33 West 39th St., New York, N. Y.

American Society of Heating and Ventilating Engineers — Annual meeting, Biltmore Hotel, New York, N. Y., Feb. 5-8. A. V. Hutchinson, secretary, 51 Madison Ave., New York, N. Y.

American Road Builders Association — Annual convention, Hotel Stevens, Chicago, Ill., Jan. 22-26. Charles H. Upham, secretary, 938 National Press Bldg., Washington, D. C.

EXHIBITIONS

National Alliance of Art and Industry—Industrial Arts Exposition, R.C.A. Bldg., New York, N. Y., Feb. 15-March 15. Alon Bement, director, 30 Rockefeller Plaza, RC.A. Bldg., New York, N. Y.

National Automobile Show — Grand Central Palace, New York, N. Y., Jan. 6-13; Coliseum, Chicago, Ill., Jan. 27-Feb. 3. Alfred Reeves, manager, 366 Madison Ave., New York, N. Y.

Exhibition of Heating and Ventilating Equipment — Grand Central Palace, New York, N. Y., Feb. 5-9. A. V. Hutchinson, secretary, 51 Madison Ave., New York, N. Y.

NEW MATERIALS and PARTS

Trent Electric Heating Cartridge Units

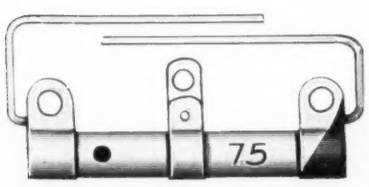
For concentrating heat in process machines, for platens, glue and compound vessels, soldering irons, dies. Cartridge type heating units are available to fit in holes $\frac{3}{8}$-in. to $1\frac{5}{16}$-in. diameter, in standard lengths from $1\frac{3}{4}$-in. up to 24-in. Casings are made of brass, but steel and stainless steel can be supplied. Closed ends are sealed by brazing. Available with twin pigtails, separate pigtail or spade terminals, for 115 or 230 volts. Also available as a hollow water-tight unit, designed to slide over a mandrel or tube. This type delivers heat on both inside and outside diameters. Also supplied as a coiled immersion unit with casings of copper, nickel, steel or stainless steel. Wire spacing and refractory construction designed to produce low thermal lag. Harold E. Trent Co., 618-640 North 54th St., Philadelphia, Pa.

Westinghouse Arc Arrester

Designed for mounting on the base of a contactor or line starter. Consists of a small capacitor and resistor connected across the poles on the load side of the contacting device. Is said to speed up a.c. arc extinctions without the use of mechanical devices to suppress the arc, and is said to be so effective that less than one-half cycle arcing occurs with the arc arrester, as compared to a persistence of five to eight half cycles under otherwise identical conditions without it. Designed so that but one unit is required for a three-phase contactor. On three-phase reversing service one arrester acts to relieve either forward or reversing contactor. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Ohmite Center-Tapped Resistors

Similar in construction to the Wirewatt resistors with the addition of a center-terminal lug. Resistance wire is wound over a porcelain core and covered with an insulating material. Lugs and leads are tinned. Resistors are color-coded and are also marked with the resistance value. Rated at 1 watt, and available with resistance values of 10, 20, 25, 50, 75, 100 and 200 ohms. Ohmite Mfg. Co., 636 N. Albany Ave., Chicago, Ill.

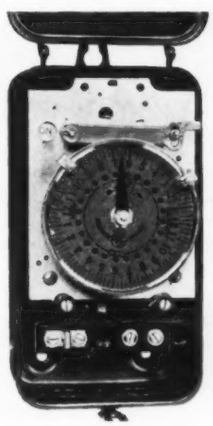


Speedized Wood Screws

Are chemically treated steel screws said to require one-third the power needed to drive plain wood screws. Available in all standard sizes of plain wood screws. Elco Tool & Screw Corp., 1800 Broadway, Rockford, Ill.

General Electric Time Switch

Driven by a Telechron motor, uses mercury-to-mercury switches. The adjustable riders give two operations every 24 hours, with provision for installing additional riders. Case supplied with or without window. Finished in black lacquer, it is designed for indoor service, and therefore has no cover gasket. Mercury switch available in several forms for 115 or 230 volts with a capacity of 30 amp. Designated as Type T-15, case dimensions are $10\frac{1}{16}$-in. high, $5\frac{3}{8}$-in. wide, $4\frac{11}{16}$-in. deep, weighs $6\frac{1}{2}$-lb. General Electric Co., Schnectady, N. Y.

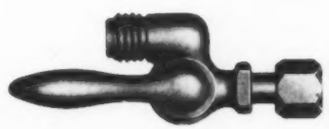


Abrasion-Resisting Cast Iron

Said to have from 2 to 12 times longer life in handling coke, stone, ore and other abrasive agents. Is a martensitic nickel-chromium cast-iron combining wear resistance with strength and toughness. Chilled surfaces are said to show a hardness of 150 to 250 numbers higher than plain chilled cast iron. When sand cast a hardness of 475 to 625 with a solid white structure is obtained. Heat-treatable for applications involving impact or alternating stresses. Not machinable in the fully hardened condition. Available in replacement parts for standard equipment and offered for general jobbing work. Robins Conveying Belt Co., 15 Park Row, New York, N. Y.

DeVilbiss Fluid Valve

For use with spray-guns, permits the operator to drain the gun of fluid or blow out any obstruction in the fluid tip without disconnecting the gun from the hose. Is a three-way cut-off valve constructed so that when the lever is turned to the rear a port in the side of the valve is opened as the fluid

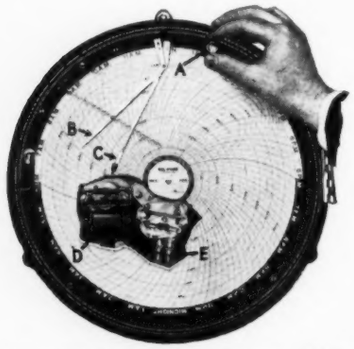


passage is closed, providing a free air passage through the valve into the fluid inlet of the gun. So designed that regardless of the position of the lever the fluid hose cannot drain through the side port. Designated as type H-5135. The DeVilbiss Co., Toledo, Ohio.

Brown Controller and Gage

For control of pressure, liquid level, or temperature, mercury switch make-and-break contacts controlled by the pen arm are engaged every six seconds by a motor-driven control table. Measuring mechanism is free of control mechanism, and making of control contacts does not depend upon friction. Mercury switch capacity is 15 amp. at 110 volts, 10 amp. at 220 volts. Control mechanism adaptable to one, two, or three contact control systems, each control mechanism independent of the other. Controller furnished in 10-in. and 14-in. circular dust-proof cases, with electric chart drive, back or bottom connected, and interchangeable for flush or wall mounting. Also supplied as a time-temperature cycle controller with a transparent cam

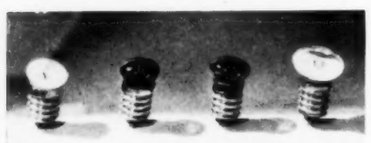
placed over the chart to shift the control index. The Brown Instrument Co., Philadelphia, Pa.



Records and Controls Temperatures up to 1200° F.
(a) Knob for changing setting of B. (b) Index showing temperature at which control is set (c) Temperature pen. (d) Electric motor selects switch position, depending upon relation of C to B. (e) Mercury in glass switches, capacity 15 amperes at 110 volts, positive in action, are mechanically positioned by motor D. No power required from pen. Eliminate relays in most applications.

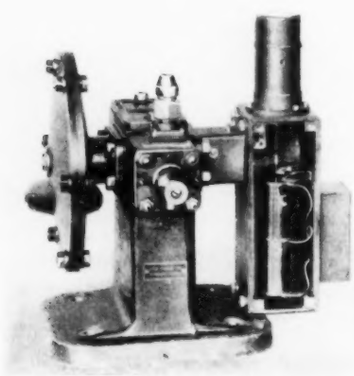
G.E. Toy Lamps

Available for one-cell or two-cell toys in two bulb sizes with enamel back and with transparent red and green bulbs. Current consumption for the one-cell type is 0.3 amp., and 0.22 amp. for the two-cell type.



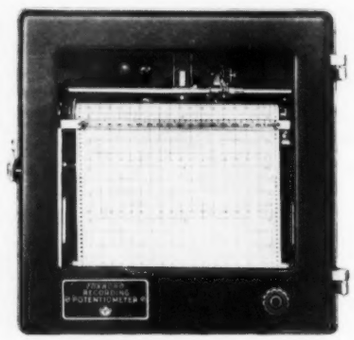
Askania Proportional Control

For the control of relative flows such as required for combustion. Illustration shows a regulator for heavy fuel oil and air using impulses from a Brown flow meter. From flow meters in the two lines impulses are transmitted to opposite sides of the diaphragm, and the differential pressure operates a control for one of the quantities. Regulator can be provided with impulse system for liquids, gases or steam. Possible also to proportion flow to temperature or pressure. While one regulator can handle only two impulses, regulators can be used in combination when three or more flows are to be controlled. H. A. Brassert & Co., 310 S. Michigan Ave., Chicago, Ill.



Foxboro Recording Pyrometer

Potentiometer type temperature recorder is housed in a fume-tight and dust-proof case, designed for either



flush or surface mounting, with aluminum cover having a large window. Mechanism arranged so that it can be swung forward out of the case, giving accessibility to the drive motor. The chart is 12-in. wide, with 10 in. exposed. Single-record instruments have a recording pen and balancing action occurs every five seconds. Multiple-record instrument prints as many as six different colored square record dots for different temperature records on one chart with a record every 45 seconds. Drive motor is mounted in rubber bushings and is completely inclosed, running in ball bearings. Scale plate is transparent, unbreakable. Terminal chamber sealed from the interior of the case, is accessible without opening the door. The Foxboro Co., Foxboro, Mass.

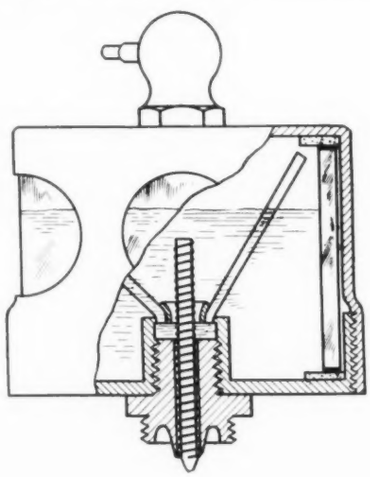
Patterson Unipower Agitators

Available in both vertical and horizontal types with shaft speeds as low as 1 r.p.m. when using an 1800 r.p.m. motor. Available in sizes of 40 hp. down to fractional sizes in several frame types, requiring no additional bearings or flexible couplings for installation. Consists of a ball bearing vertical motor, capable of withstanding 400 per cent momentary overloads, a specially developed speed reducer unit running in oil, and a one-piece frame or housing. Guide bearing in bed-plate is oilless and takes side

thrust. Working parts are completely inclosed and run in oil. The Patterson Foundry & Machine Co., East Liverpool, Ohio.

Hallanger Oiler

Designed to provide a constant flow, protects the oil from dust and dirt. Oil feeds from 1 to 5 drops per min., and as low as 1 drop in 4 to 5 min. are available in designs using metering pins with various wire springs. By selecting springs with various distances between the turns of the spring, the oiler is built to provide rates of discharge covering a wide range. Oiler holds sufficient quantity for one day. Hallanager Oiler Sales, 630 St. Jean Ave., Detroit, Mich.



Two New Hygrade Lamps

A 100-watt lamp of special construction is designed to withstand high-frequency vibration for applications where standard 100-watt lamps give poor service. Filament and support-

ing members are mounted in a flexible structure. Filament wire is specially designed and held by six filament supports attached to the glass arbor. Filament leads are flexible, and arbor is supported on a steel wire spring. The initial light output is said to be the same as the standard 100-watt lamp, with average light output twelve per cent less than standard.

Another new product of this company is a white bowl bulb constructed with the lower half of the glass envelope finished with an enamel-like coating sprayed on the inside of the bulb. Manufacturer states that the enamel coating is prepared so that no discoloration occurs. Hygrade Sylvania Corp., 500 Fifth Ave., New York, N. Y.

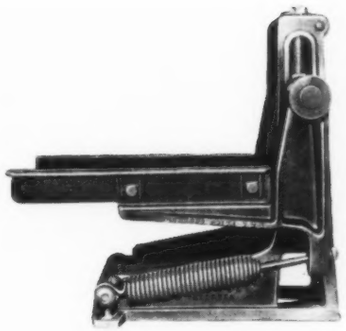
Multiple-Disk Magnetic Clutch

For operation in oil, water and other liquids in applications requiring smooth acceleration, starting without jerk or jar, and where automatic electric control is desirable. Available in sizes ranging from 6-in. diameter. Clutches are built with from 2 to 8 disks; coils are impregnated against

moisture; collector rings, brush holders and brushes are supplied as part of the clutch. One point adjustment is provided to take up lining wear. Magnetic Mfg. Co., Milwaukee, Wis.

Rockwood Vertical Drive Motor Base

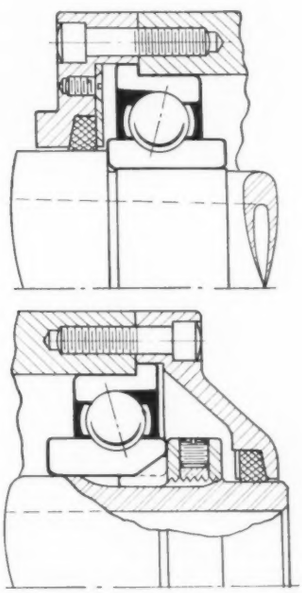
Designed to provide for and maintain correct belt tension. Motor is mounted on two adjustable horizontal arms suspended from a pivot shaft. When motor is located above the driven pulley, springs are adjusted to support the motor weight in addition to the necessary belt tension. When the motor is located below the driven pulley, the springs support only the portion of motor weight not needed for belt tension. Adjusting screws are provided to move the pivot shaft vertically to adjust for elongation of belt



in service. This adjustment also provides means for aligning the two pulleys so that the belt will track. Base designed so that belt tension can be readily adjusted to operate under widely varying loads by turning the nuts on the ends of the coil-spring rods. Rockwood Mfg. Co., Indianapolis, Ind.

New Departure Flanged Ball Bearings

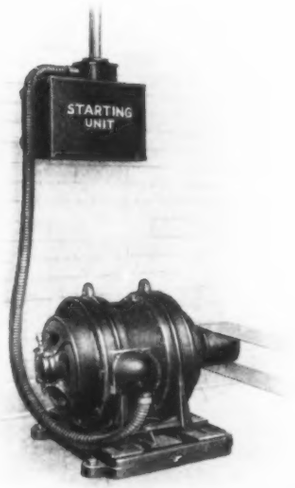
Line of precision ball bearings embodying a flange on the outer race, making it possible to bore housing straight through at one set-up. The bearings are separable for quick and easy assembly. Inner rings are extra wide to provide non-deflecting seat



and are key-slotted to prevent creeping about the shaft. Preload is applied through the inner rings. Bearings supplied in sizes from 30 mm. up to 130 mm.. to precision specifications, and are obtainable in two bore sizes for each standard outside diameter. New Departure Mfg. Co., Bristol, Conn.

Marble-Card Capacitor Motors

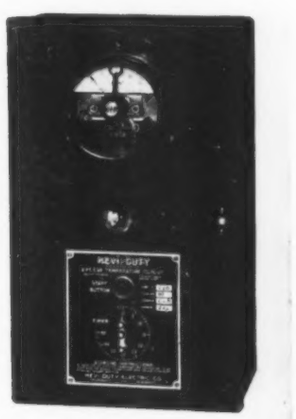
Similar in construction to a two-phase squirrel cage motor with a condenser connected across one phase for starting. Condenser and contactor are mounted in the starting unit. Grease lubricated ball bearings are standard equipment. Starting capacitor is electrolytic type, giving starting torque of approximately 200 per cent of full



load torque. Available in N.E.M.A. frame and short length M-C frames, in sizes from 1 to 15 hp., nominal speeds from 600 to 3,600 r.p.m. Marble-Card Electric Co., Gladstone, Mich.

Hevi-Duty Excess Temperature Cut-Out

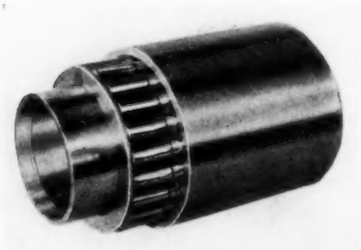
For gas, oil and electric heating apparatus, protects against heating a charge beyond a predetermined temperature. An adjustable target is set at the desired maximum temperature, and a relay located in the heating chamber operates a contactor controlling the main heating circuit when



this temperature is reached. A pilot lamp indicates whether the circuit is on or off. Hevi-Duty Electric Co., Milwaukee, Wis.

Fafnir Roller Bearing

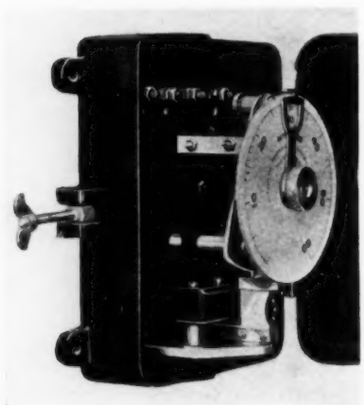
Consists of inner and outer rings with a roller and cage assembly. Hardened rolls are of high carbon chrome steel, rings are of alloy steel said to have a high fatigue value. Cage is carburized and end rings are hardened and rivetted in place. The use of spacer bars instead of rivets through



the roller is said to result in a stronger roller having a higher load capacity. To allow for expansion of the inner ring when shrunk to the shaft, roller assembly is made with a fixed radial clearance, which also allows for some expansion caused by heating. Available for shaft diameters from 3½-in. to 7⅞-in. Fafnir Bearing Co., New Britain, Conn.

Adjustable Timing Contactor

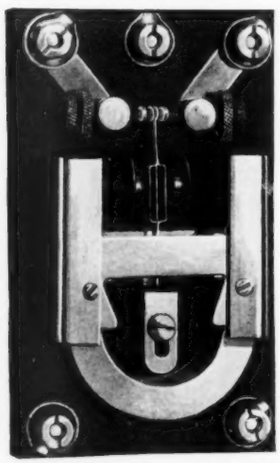
Designed to provide a momentary contact at the end of an adjustable time interval. Also available for closing the load circuit at start of the time cycle, breaking the circuit when a specified time interval has elapsed, or making and holding the load circuit only at the expiration of the time set. Contact arm driven by a reversing synchronous motor, moves alternately



clockwise and counter-clockwise within the time limit set. Contacts rated at ten amp. 110 volts, a.c., are said to be unaffected by vibration or level. Designated as Type 1290, with a sheet steel case and hinged glass front and is 7x5½x4¼ in. Also available with a cast aluminum case and gasket sealed lid. Automatic Temperature Control Co., Inc., 34 East Logan St., Philadelphia, Pa.

Dunco Polarized Relay

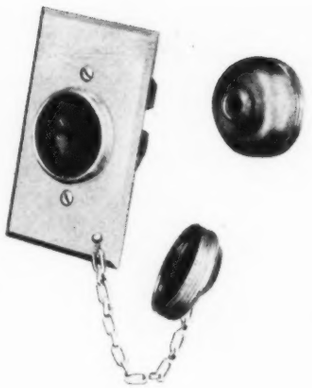
Designed for operation on d.c. power where a reversal of current in the operating coils reverses the contacts, no contact being made when the coils are deenergizd. Consists of a permanent magnet and two coils arranged to move an armature carrying moving contacts. Strength of permanent magnet is adjusted by a shunt, while position and tension on the moving contact



member as well as the fixed contacts are also adjustable. Mounted on an insulating base 5x3x½ in., relay is 2½-in. high. Contacts rated at 2 amp. 110 volts, a.c. or ¼ amp., 115 volts, d.c., non-inductive load. Available with total coil resistance as high as 6,000 ohms, operating on power as low as 0.01 watt, d.c. Supplied with sheet metal, or cast aluminum weatherproof housings, or glass covers. Struthers Dunn, Inc., 146 North Juniper St., Philadelphia, Pa.

Weatherproof Outlet

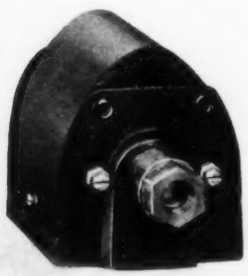
For use with lighting fixtures and appliances exposed to weather. Made of brass and cadmium coated to resist rust and corrosion. Plate has a solid rubber pad attached to the back to insure a snug fit when the plate is screwed in place. Cap shown at the right slips over the cord plug and screws into the outlet. Rubber insulated cap, chained to the flush plate of



the outlet covers the opening when not in use. General Electric Co., Schenectady, N. Y.

Westinghouse Fluid Valve

Is an electrically operated check valve for handling compressed air and liquids by remote control. As a whistle valve, the coil can be energized through a pushbutton at any convenient location. As a safety valve in an airplane gasoline tank line, the valve can be connected in the ignition circuit, cutting off the supply of gasoline when the ignition switch is open.



Said to be suitable for hot or cold water, oil and gases. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Planigressive Multi-Speed Reducer

Built in two, three or four specified speed reductions with a range of 2.66 to 1 up to 20 to 1 ratio coupled with an additional speed reducer. Planetary construction allows all gears to remain in mesh; can be shifted to any speed while running under load, running idle or standing idle. Unit is made integral with motor, or with an input shaft independent of the prime mover. Splash-lubricated by planet gears dipping in oil reservoir, with oil seals to prevent leakage at the input and output shaft.

Available in sizes from ¾ hp. to 10 hp., either horizontal, vertical or right angle drive. Also supplied with explosion-proof or drip-proof motors. Composition gears run on bearings supported at both sides in a one-piece planet cage supported on both ends by ball bearings. An adjustable overload clutch is built into the reducer to take care of shock loads. The Planigressive Reducer Sales Co., Milwaukee,, Wis.

FOR THE ENGINEERING DEPARTMENT

Tasco Slide Rule

Circular slide rule of nickel silver with etched divisions is 2¾-in. in diameter, weighs 3½-oz., has five scales on the front side: Log, C, D, CI or C inverted, and K scale. Reverse side has sine and tangent scales and an A and D scale for finding squares and square roots. Celluloid riders on each side carry hairlines. Length of multiplication-division scale is 6.3-in. as compared with five inches on A scale of regular 10-in. slide rule. An improved etching process is said to provide an average accuracy of ¼ of 1 per cent, with the added advantage that the accuracy is unaffected by moisture absorption or changes in temperature. Supplied with a manual giving instructions for its use. Tavella Sales Co., 25 West Broadway, New York, N. Y.

Rawlplug Gage

Designed for quick and accurate selection of the proper Rawlplug for a given wood or lag screw. Is 6-in. long 1⅝-in. wide. Tapered slot at each end is designed so that the size needed is indicated by inserting a wood or lag screw between the jaws. One edge is graduated in sixteenths, and the other edge forms a 5-in. rule with a slot for accurate measurement of length of counter-sunk head screws. The Rawlings Co., Inc., 98 Lafayette St., New York, N. Y.

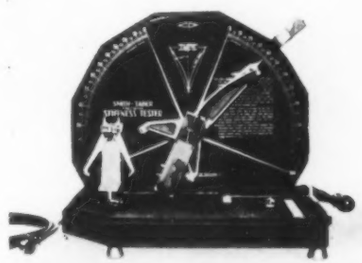
Profile Projector

Consisting of a vertical projector, a supporting table 2⅜-in. diameter, and a steel cabinet containing an inclined mirror, and a screen 12½x15¾ in. set at an incline in the top of the cabinet. Designed so that shadows of the operator's head or hands do not interfere with work at the screen. Projected imagine can be accurately measured, copied or photographed. Glass plate supporting table is set in a circular ring permitting rotation of the object, while two cross slides give movement in two horizontal coordinates. Table can also be raised and lowered for focusing. Supplied with four magnifications of 10, 20, 50 and 100 diameters, guaranteed accurate to 1 part in 2,000. Magnification readily changed by changing objectives. Green and blue filters also supplied. Several accessories provide supports for various types of articles on supporting table. Steel columns support a framework and canopy large enough to permit several persons to examine the projection at once. Overall dimensions are 45x48 in., 75-in. high. The R. Y. Ferner Co., 1008 Fifteenth St., N.W., Washington, D. C.

A practical application of the profile projector, checking the projected image of a gear and pinion

Smith-Taber Stiffness Tester

For testing wire and strip, metal foil, filament, Bakelite sheet, textiles, imitation leather, paper, fiber, pressboard. Strip cutter, gaging both length and width, is built into the device. Test strips are clamped in upper part of pendulum pivoted at the center of the scale. Indicator arm is moved



across the scale until pendulum is lifted from contact and lamp on the platform is lighted. Operation is repeated over the opposite scale, the total reading representing the stiffness of the specimen. Sensitive attachment shown at the left of the platform is used for more pliable materials. The instrument is claimed to accurately test materials to which hardness tests cannot be applied. Sensitivity of the tester is said to give accurate measurements of plastic flow, and the effect of humidity on paper, as well as the pliability of fabrics. The pendulum is suspended on double ball bearings;

tungsten steel points make the electrical contact. Smith-Taber, North Tonawanda, N. Y.

Stanley Flexible-Rigid Steel Rules

Supplied with a round brass and steel case, with two-tone chromium-plated finish, has a removable blade graduated on both sides on all four edges. Blades are nickel-plated with a safe pull-push blade action. Cases are 2-in. diameter. Arrow on case gives inside measurements when end of blade and bottom edge of case butt against work. Stanley Tools, New Britain, Conn.

NEW BOOKS and PUBLICATIONS

1933 Book of A.S.T.M. Standards

Two Volumes. Part I, Metals, 1,002 pages. Part II, Non-Metals, 1,928 pages. 6x9 in. Blue cloth board covers. Published by American Society for Testing Materials, 1315 Spruce St., Philadelphia, Pa. Price, each part, $7.50; complete book, $14.

Published every three years, this 1933 edition contains all the standard specifications, methods of testing, definitions and recommended practices adopted by the Society. Part I includes 184 standards, 104 applying to ferrous metals, 70 to non-ferrous metals and 11 on testing methods. Part II contains 285 standards covering such materials as petroleum and rubber products, cement, refractories, coal, paints and textiles. Each part is provided with two indexes, one in numerical sequence and the other a complete subject index.

The American Railroad in Laboratory

544 pages, 6x9 in. Heavily illustrated. Paper covers. Published by American Railway Association, Washington, D. C. Price 50 cents.

Prepared for the general public as well as for designers of railway equipment, this summary shows the extent of the work of the railroads on research and development problems. Of the 272 active research projects listed, more than two-thirds pertain to locomotives, freight cars and engineering.

Following a brief history of railroad development, research projects are classified and summarized in separate chapters on such subjects as locomotives and cars, roadway and structures, signalling, automatic train control and freight containers.

Internal-Combustion Engines

Robert L. Streeter and Lester C. Lichty. 539 pages, 6x9 in. 327 illustrations. Green cloth board covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $4.50.

Revised and amplified, this fourth edition includes new material on cycles, fuels, detonation, carburetion, fuel injection, manifolding, cooling, lubrication and design. As in previous editions theory, analysis and design of internal-combustion engines are all given in well-illustrated textbook form. Such subjects as flame travel, combustion-chamber design and synchronous vibrations are included, as well as design data and information on stresses, moments, sections and materials. Conventional design of principal engine parts is treated in the closing chapter.

Electrons at Work

C. R. Underhill. 354 pages, 6x9 in. 220 illustrations. Clothboard covers. Published by the McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $3.

Using a textbook style but a general treatment, the author attempts to popularize the electronic theory and the uses of electronic devices. Two chapters on gaseous-discharge tubes, and light and color, and four chapters on photo-electric cells and devices are accurate, interesting and fairly well illustrated, and will be of interest to the designer engaged in applying electronic equipment. In the opinion of the reviewer, a more detailed treatment of typical applications giving specific data as to capacity, limitations and comparative performance of electronic devices would add much to the value of the text from the designer's viewpoint.

▼

Research Bulletin

Engineering Experiment Station, University of Illinois, Urbana, Ill. Bulletin No. 257, "The Friction of Railway Brake Shoes," by Edward C. Schmidt and Herman J. Schraeder. In this 100-page report is a summary of an investigation using two kinds of brake shoes on both chilled iron and steel wheels to determine values of coefficient of friction at various speeds over the usual range of shoe pressures. Tests were made simulating conditions under which a train is brought to rest. Also, the A.R.A. standard tests for brake shoe wear were made. Definite and regular decreases in the coefficient of shoe friction were found as the speed increased, the coefficient decreasing as the pressure increased. Chilled wheels gave a higher coefficient than was obtained on steel wheels.

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Aluminum—Aluminum Company of America, Pittsburgh, Pa. Booklet, 5x8 in., 235 pages, "Aluminum in Architecture." Information on the characteristics of aluminum and aluminum alloys suitable for the building industry. Data on designing, fabricating and finishing as well as protection, erection and maintenance. Included is a section of 30 pages of tabular data, and a chapter on specifications. Applications and design details are well illustrated.

Bearings — Kingsbury Machine Works, Inc., 4324 Tackawanna St., Philadelphia, Pa. Bulletin G-1, 58 pages, "Marine Type and Heavy Industrial Mountings," describes the principle of Kingsbury horizontal thrust and journal bearings. Well illustrated applications and design details, as well as data on capacities, lubrication requirements and catalog information on various types of mountings.

Couplings—Link-Belt Company, 910 South Michigan Ave., Chicago, Ill. Booklet, 14 pages, 6x9 in., "Link-Belt Flexible Couplings" with catalog and design data on several types of flexible couplings.

Metal Coatings—Metals Coating Co. of America, 495 Third St., Philadelphia, Pa. Bulletin 1201, 8 pages, "Metalayer Process and Equipment," with data on procedure and properties of sprayed metal coatings.

Motors — Crocker-Wheeler Electric Mfg. Co., Ampere, N. J. Twelve-page booklet, "Crocker-Wheeler Products," with illustrations of a.c. and d.c. motors for various applications .

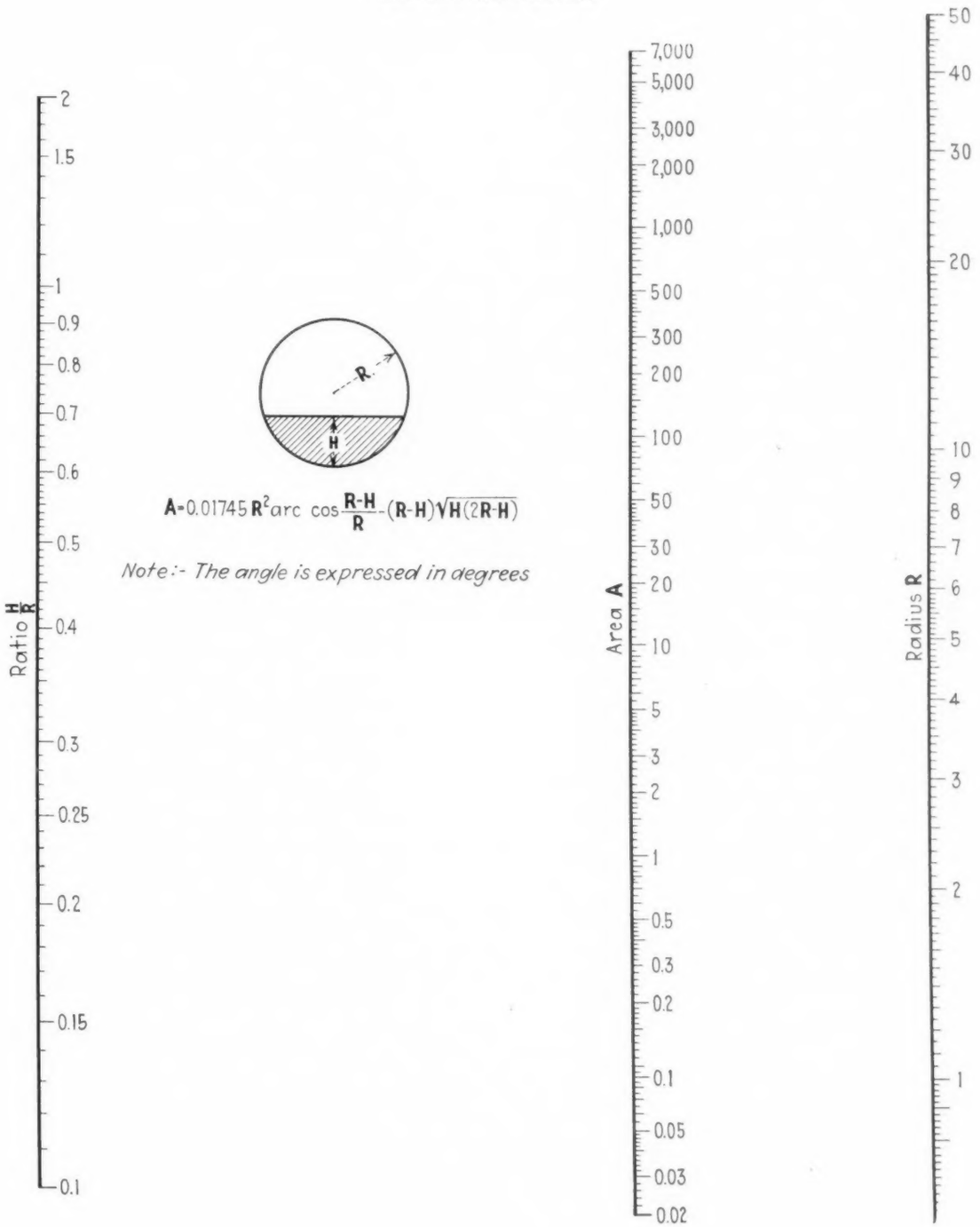
Plastics—Synthane Corp., Oaks, Pa. Chart of characteristics, working load and design data for Synthane laminated Bakelite gear material.

Steel — Carnegie Steel Co., Pittsburgh, Pa. Supplement to "Pocket Companion," of 24 pages on CB Sections with elements of sections, dimensions and design data.

Zinc—The New Jersey Zinc Co., 160 Front St., New York, N. Y. Booklet, 20 pages, "Design for Profit," gives data on zinc die-castings and a number of case histories in die-cast design.

Areas of Circular Segments

M. G. Van Voorhis



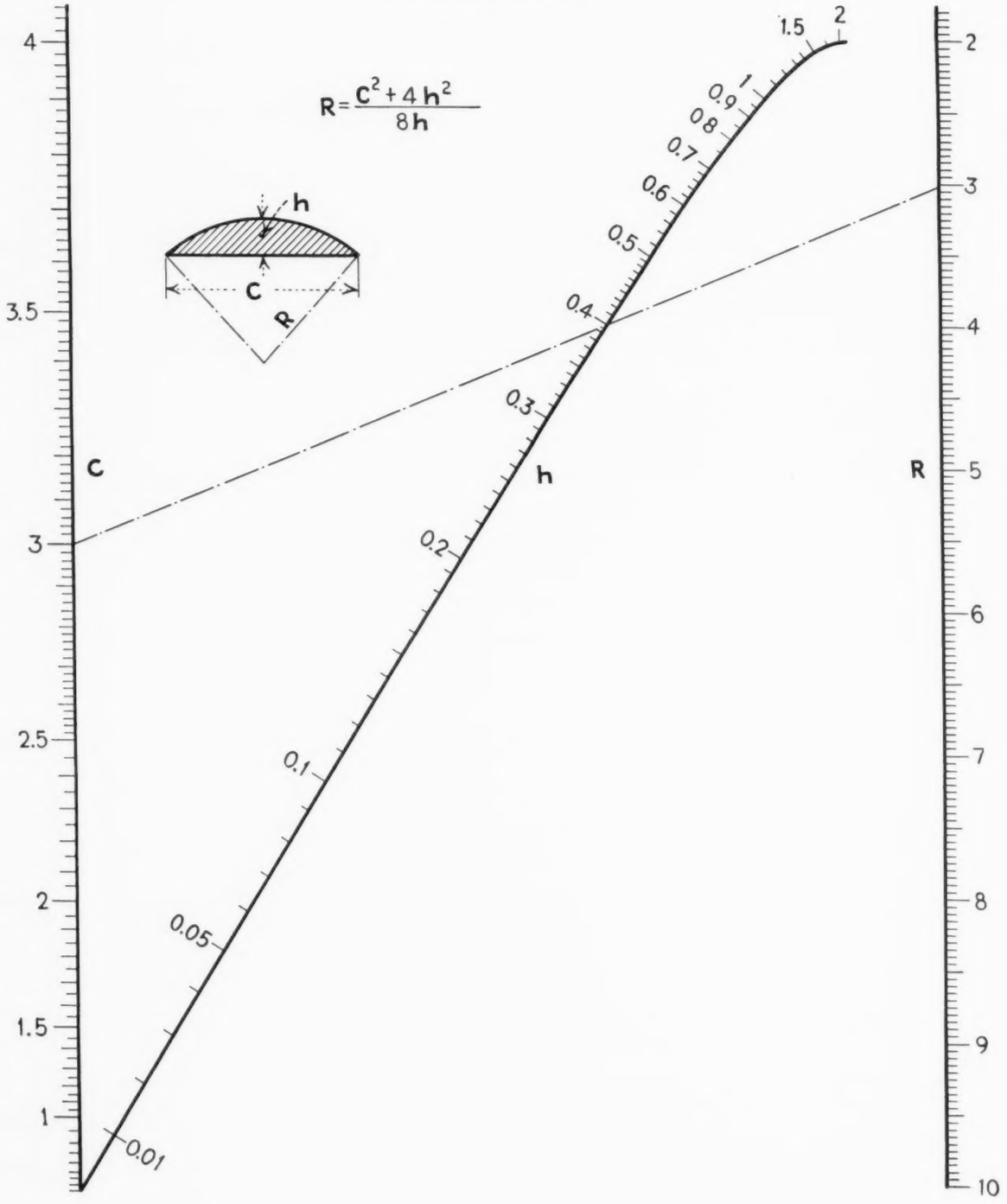
$$A = 0.01745\,R^2 \arccos\frac{R-H}{R} - (R-H)\sqrt{H(2R-H)}$$

Note:- The angle is expressed in degrees

PRODUCT ENGINEERING © REFERENCE BOOK SHEET

Chordal Heights of Circular Segments

M. G. Van Voorhis



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